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# Acoustic Design of a Public Space Using Perforated Panel Resonators

**School of Electrical Engineering**

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ABSTRACT OF THE
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Acoustic conditions in a public spaces are not always paid attention to. When the usual construction materials used in such places are acoustically reflective, this combination is detrimental for the acoustics. Therefore the reverberation time is high and acoustical conditions are weak at these type of places.
Sometimes the acoustical treatment can be left out because it spoils the visual appearance of the space. Therefore in this thesis is presented a case study of an acoustic treatment in a public space located in Espoo, Finland. The space is a food court area designed to an extension of a Finnish shopping center called Iso-Omena.
The construction is still on its design phase, so the only possibility was to use a 3D model for computer simulations in order to investigate the acoustical conditions in the space. The acoustical treatment in the space was chosen to be a perforated panel resonator, because using this the appearance of the surface materials would remain as similar as is the original design. Different resonator absorbers were modeled and from these the best alternative was chosen, which was then implemented in the 3D model for the simulations. For the resonator, two different panel thicknesses were available, and both of these were considered.
Based on the results of the computer simulation, the reverberation time of the space reduced roughly 0.5 seconds at 500 Hz with a perforated panel resonator installed at the walls. The structure of the resonator was 5 mm sheet metal with 7 % of perforations backed by a 50 mm mineral wool and 95 mm air. With small acoustical designing it was able to lower the reverberation time to an acceptable level, and still preserve the original visual appearance of the space.

Keywords: Acoustics, absorption, resonance, reverberation, simulation

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Julkisten tilojen akustisiin olosuhteisiin ei aina kiinnitetä riittävästi huomiota. Kun tämä yhdistetään akustisesti koviin rakennusmateriaaleihin, joita näissä tiloissa usein käytetään, tilanne on akustiikan kannalta epäsuotuisa. Tämän vuoksi jälkikaiunta-aika tilassa on korkea ja akustiset olosuhteet heikot.
Akustinen käsittely saatetaan jopa jättää kokonaan pois jos se pilaa tilan visuaalisen ilmeen. Sen takia tässä työssä esitellään Espoossa sijaitsevan julkisen tilan akustiikan tapaustutkimus. Tila on ravintolamaailma, joka sijaitsee Iso-Omenan ostoskeskuksen laajennuksessa.
Koska rakennus on vasta suunnitteluvaiheessa, ainoa mahdollisuus tutkia tilan akustiikkaa on 3D-mallinnus ja simulointi. Tilan akustiseen käsittelyyn valittiin rei'itetty paneeli, joka toimii resonaattorina, koska näin tilan pintamateriaalia ei tarvitse muuttaa ja visuaalinen ilme vastaisi alkuperäistä suunnittelua. Erilaisia resonaattorivaihtoehtoja mallinnettiin ja näistä paras vaihtoehto mallinnettiin 3D-mallin avulla. Resonaattoria varten oli saatavilla kaksi eri pintamateriaalia, joiden paksuus erosi toisistaan, ja näiden molempien käyttöä tutkittiin.
Mallinnuksen tulosten perusteella jälkikaiunta-aikaa saatiin laskettua tilassa n. 0.5 sekuntia 500 Hz oktaavikaistalla käyttäen reikälevyresonaattoria seinissä. Resonaattorin rakenne oli 5 mm metallilevy, jossa oli 7 % reikiä, ja levyn takana oli 50 mm mineraalivillaa ja 95 mm ilmaväli. Tällä verraten pienellä akustisella toimenpiteellä jälkikaiunta-aikaa saatiin laskettua hyväksyttävälle tasolle ja samalla pystyttiin säilyttämään tilan alkuperäinen visuaalinen ilme.

Avainsanat: akustiikka, absorptio, kaiunta, resonanssi, mallinnus

# Preface

This thesis was written in and funded by Ramboll Finland Oy as a part of the project of the design of Matinkylä's Metro station in Espoo, Finland. I would like to express my gratitude to Ramboll for providing me with the opportunity to write my Master's thesis as a part of this interesting project. A thank you goes also to the staff at Iso-Omena for letting us to do reverberation time measurements in the building.

I also want to thank the supervisor of the thesis, Prof. Vesa Välimäki for his invaluable advices, instruction and discussions throughout the writing process.

The advisor of this thesis, Petteri Laine, has also provided me with a lot of help and discussions about the computer simulations, reverberation time measurements and, in effect, every aspect of this thesis. Many thanks for the knowledge and help.

Finally I want to thank my wife Enni, who has been there for me every day throughout the writing process and during my whole studies.

Helsinki, 19.1.2014

Miikka Valtonen

# Contents

# Symbols and Abbreviations

## Symbols

| | |
|---|---|
| $c$ | Speed of sound in air |
| $d_\psi$ | Diffusion coefficient |
| $f$ | Frequency |
| $L_A$ | A-weighted sound pressure level |
| $L_I$ | Sound intensity level |
| $L_p$ | Sound pressure level |
| $L_W$ | Sound power level |
| $R$ | Sound insulation |
| $s$ | Scattering coefficient |
| $T_{20}$ | Reverberation time (-5 – -25 dB range) |
| $T_{30}$ | Reverberation time (-5 – -35 dB range) |
| $T_{60}$ | Reverberation time |
| $\alpha$ | Absorption coefficient |
| $\varepsilon$ | Porosity |
| $\eta$ | Viscosity coefficient |
| $\rho$ | Density of a material |
| $\sigma$ | Flow resistivity |
| $\tau$ | Penetration ratio |

## Abbreviations

| | |
|---|---|
| HVAC | Heating, Ventilation & Air Conditioning |
| LEED | Leadership in Energy and Environmental Design |
| MDF | Medium Density Fiberboard |
| MPP | Microperforated Panel |
| QRD | Quadratic Residue Diffuser |

# 1 Introduction

The construction of public spaces that have a large atrium or other space with a high volume, such as large shopping centers, has increased in Finland in the recent years. The largest investor in shopping centers in Finland alone has built three new centers and renovated and extented five existing centers in the $21^{st}$ century. New schools are also built and old ones are being refurbished and modernized. In these there are usually a gym class that has a high volume. Large office complexes have also an atrium with high ceilings made out of glass, which affords natural lighting. However, the acoustic conditions in these places are not always satisfactory. This is due to the fact that what is usually visually pleasing may be acoustically detrimental. Proper attention is not paid to the acoustic conditions of these spaces in the design phase, and afterwards the changes may be too expensive to conduct. Furthermore, there are no strict regulations for the acoustical conditions in such places, only a few guidelines concerning the reverberation time.

One contributing factor to long reverberation times and unclear and loud sound scapes are hard, acoustically reflecting surface materials, e.g. glass, concrete, brick and marble, which are used together with little or no absorptive material. This creates an uncomfortable environment to spend time in. For a company, bad acoustics in an atrium gives poor first impression to visitors and clients. At a shopping center, more often they are not only places to quickly do your everyday grocery shoppings, but more and more time are spent in the centers. In many centers there are much more than clothing or electronics stores, including restaurants, bowling alleys, movie theaters and even spas. Long reverberation time lowers speech intelligibility and may even prevent people hearing announcements from the PA-system. In worst-case scenario, in the case of fire or other hazard, evacuation calls and directions may be unintelligible. Therefore the acoustics of these spaces should be carefully considered to make the stay in the area pleasant, enjoyable and safe.

In addition to nuisance, hearing losses may be caused by even moderately elevated sound levels. The Finnish residential health instructions (Asumisterveysohje) states that an average sound level of $L_{A,eq,24h}$ = 65–75 dB may contribute to a temporary hearing loss. In a study conducted in Ankara, Turkey (Döcmeci and Yilmazer, 2013), a measured average sound level at a shopping center was reported to be 63–68 dB. Because long reverberation time increases the overall sound pressure level in the space, without proper acoustic designing, the noise levels in shopping centers may be a contributing factor to a temporary hearing loss.

This thesis introduces a case study for acoustic design of a food court area in an extension of a popular shopping center in Espoo, Finland. The existing shopping center, Iso-Omena, is extended and renovated, and an inside food court area is planned to the extension wing. The surface materials are planned to be mostly glass and concrete. Without any acoustical treatment the space is assumed to be very noisy and clamorous due to a long reverberation time. Therefore a 3D-model of the space is created, and with that computational solutions to lower the reverberation time to an acceptable level are researched. This includes modeling of a perforated panel resonators, which can be tuned at a specific frequency to absorb sound. The structure of this thesis is as follows: basic theory of sound and room acoustics are given in the second section, the third section presents theory about sound absorption, reverberation and diffusion. The fourth section consists of considerations of Finnish regulations concerning acoustics in public spaces, a review of earlier studies and the design tools and software that was used in this thesis to make the simulations. In the fifth section the results from the acoustic simulations are presented with some analysis. The sixth section concludes the thesis.

# 2 Theory of Acoustics

In this section the basic theory of sound and acoustics is given. First, a basic understanding about sound waves is given , followed by introduction of human hearing, and finally acoustical resonance is considered.

## 2.1 Sound Waves

Sound is a longitudinal wave transferred in a medium that can be either solid, liquid or gas. The speed at which the wave travels depends on the medium and its properties. In a gas, e.g. air, sound wave consists of air molecules compressing and depressing, creating pressure changes that is heard as a sound. The velocity at which the wave travels in air is defined by

$$c = 331.3 + 0.6t \; [m/s] \tag{1}$$

where $t$ is temperature of air in degrees Celsius. Using Eq. 1 it can be calculated that speed of sound in 20 °C is about 343.3 m/s. In general, the more dense the molecular structure is, sound energy is transferred more easily and as a result at greater speed. In comparison, sound travels in water, which is more dense than air, roughly at 1.410 m/s.

While the movement of air molecules create the propagating sound field, the molecules itself do not move along with the sound. Instead they simply vibrate at their places, the displacement being a few hundreth of millimeters (Everest and Pohlmann, 2009). The motion is similar to a mass-spring system vibrating up and down when it is disturbed from its equilibrium position. To take the mass-spring comparison even further, the velocity of the air particles are at maximum at the equilibrium position and zero at the point of maximum displacement, as is with the mass-spring system. This movement of the particles is called particle velocity. Sound pressure and particle velocity are linked together so that when other one is at maximum at a point in space, the other is at zero. Particle velocity is also the contributing factor to the loudness of a sound, i.e. the higher the particle velocity, the higher the overall sound pressure level. Particle velocity of a plane wave traveling through a medium at the direction of x-axis is defined as

$$\mathbf{u} = \frac{A}{\rho c} e^{j(\omega t - kx)} \tag{2}$$

where $A$ is the amplitude of the wave, $\rho$ is the density of the medium, $\omega = 2\pi f$ is the angular frequency and $k$ is the wavenumber. (Cox and D'Antonio, 2004)

The range of changes in sound power, intensity and sound pressure is very large, and using logarithmic scale and decibels alleviates the problem of very large numbers. Decibels are thus widely used in acoustics, and few of the most principle acoustic measures are introduced next. Every sound has its source, and for every source an acoustic definition for its ability to produce sound can be determined as the sound power level, which is defined as

$$L_W = 10\ \log_{10} \frac{W}{W_0} \tag{3}$$

where $W$ is the sound power of the source and $W_0 = 1\text{x}10^{-12}$ [watts] is fixed sound power reference. This is also called the sound emission of a source. Now that a sound source is defined, it is appropriate to define the sound it creates in a point in space somewhere away from the source. First a sound intensity level at a point has to be defined. That is defined as

$$L_I = 10\ \log_{10} \frac{I}{I_0} \tag{4}$$

where $I$ is the intensity in the point and $I_0 = 1\text{x}10^{-12}\ \text{W/m}^2$ is the reference intensity. Now a question rises that how do we calculate a sound intensity of a source in a point in space? To answer this question, we must have some extra information about the source and the environment it is in. A point-source (a source that radiates sound equally to all directions) in a free-field conditions is the most simple one. The intensity of the sound field is governed by $1/r^2$, where $r$ is the distance from the source. It is easier to understand this considering a growing sphere. The area of a sphere is given by $4\pi r^2$, and while the sphere expands, the same sound power is distributed to larger area, thus decreasing the intensity. Now the intensity at a distance can be expressed as

$$I = \frac{W}{4\pi r^2} \tag{5}$$

where $p$ is the sound pressure and $\rho$ is the density of air. In other words, Eq. 5 states that the sound intensity level decreases by 6 dB every time the distance doubles, because the same sound intensity is distributed to an area four times larger than the original area. This is depicted in Fig. 1, where it can be seen that with increasing distance the surface area grows very quickly.

Intensity and sound pressure are related to each other by Eq. 6

$$I = \frac{p^2}{\rho c} \tag{6}$$

where $p$ is the sound pressure level. This means that when sound pressure doubles, intensity quadruples. Since sound is air pressure changes, it is convenient to define a logarithmic measure for sound pressure level:

$$L_p = 20\ \log_{10} \frac{p}{p_0} \tag{7}$$

where $p_0$ is a fixed reference value of 20 $\mu Pa$, which equals to human hearing threshold at 1 kHz. This is the most used measure, and when you hear someone mentioning decibels when talking about sound, they most likely are talking about the sound pressure level.

From Eq. 7 two interesting points follows. First, the hearing threshold is defined as 0 dB, and therefore it is even possible to have negative sound pressure levels.

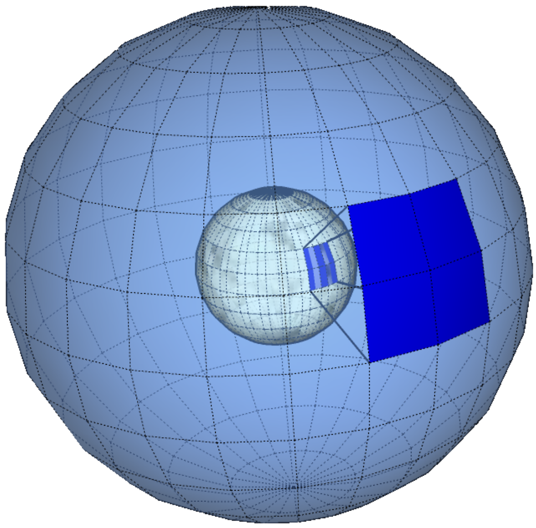

Figure 1: An illustration of a point-source emitting sound in a free-field.

Since the threshold of pain is about 120 dB, which corresponds to a sound pressure of 20 Pa, it can be seen that even the loudest sounds humans hear are very small in pressure compared to the atmospheric pressure of about 1000 kPa. Second, the doubling of sound pressure equals to 6 dB rise in sound pressure level and 20 dB increase equals 10 times higher sound pressure. From this it could easily be conducted that when two equal sound sources are on, and when the other one is switched off the sound pressure level halves and thus 6 dB decrease in sound pressure level occurs. However, the two sound sources need to be correlated for the sound pressure to double. More often sources are uncorrelated, in which case only the sound power doubles or halves, which equals to 3 dB increase or decrease in sound pressure level.

Since sound is vibration of air molecules, a frequency at which it is vibrating can be defined. The frequency of a sound wave is related to its speed and wavelength as in Eq. 8

$$f = \frac{c}{\lambda} \tag{8}$$

where $c$ is the speed of sound and $\lambda$ its wavelength. This frequency is interpreted as the pitch in which the human ear perceives sound, i.e. is the sound low or high. Human hearing system is able to perceive sounds that range roughly between 20 – 20,000 Hz, but adults may not be able to hear sounds above 10 – 12 kHz as the hearing system begins to atrophy with age (Rossing et al., 2002).

## 2.2 Human Hearing

The sensitivity of the human ear to sounds is frequency dependend. For low frequencies (under 250 Hz) the sound pressure level has to be much higher to obtain the same perception of loudness than with frequencies around 1,000–4,000 Hz. Equal loudness curves presented in Fig. 2 depicts the relative sensitivity of human hearing at different frequencies. In the picture is plotted curves in phons ranging from hearing threshold, which is 0 phons, to 100 phons. Phons are defined as being the same as the sound pressure level in decibels at 1,000 Hz frequency. From the curves it should be noted that the ear is most sensitive at frequency range of 3.5 – 4 kHz, which corresponds roughly to the first resonance of the ear canal. Furthermore, as the frequency decreases the sensitivity of the ear decreases also. The equal loudness curves are intepreted as in order to have an impression of loudness of 50 phons at 125 Hz a sound pressure level of almost 70 dB is required, compared to the 50 dB which is enough at the 1,000 Hz frequency. (Rossing et al., 2002)

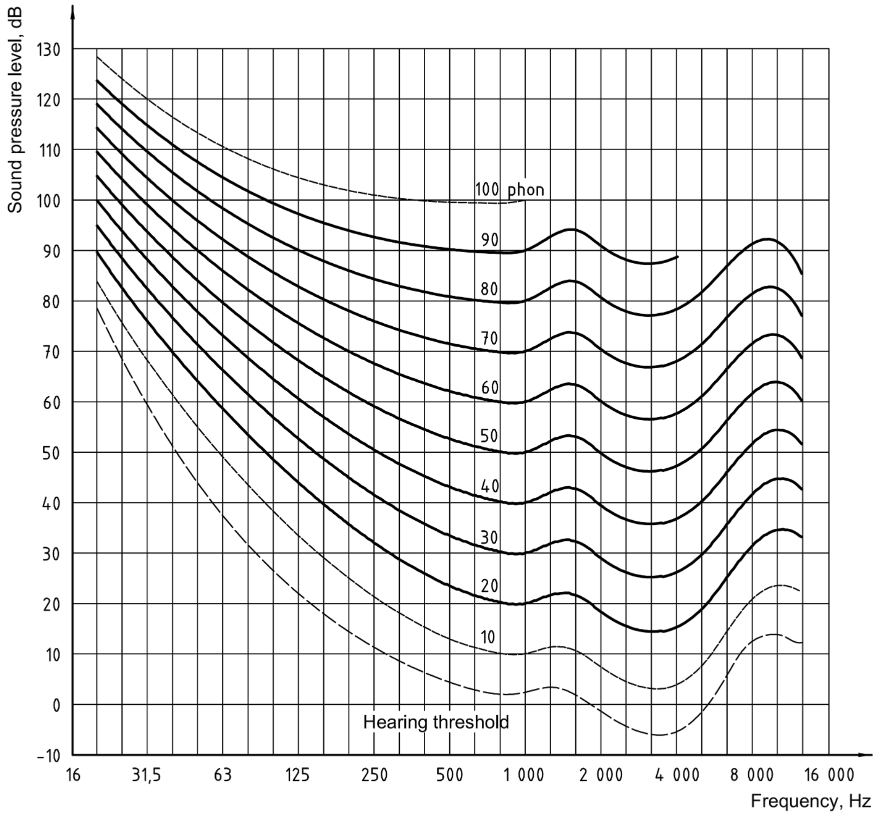


Figure 2: Equal loudness curves (ISO 226).

Now that it is noted that the ear does not hear all frequencies equally loud, it can be deduced that sound pressure calculated with Eq. 7 does not necessary correlate well with the heard impression of sound loudness. Therefore different scales have been developed for the sound pressure to correspond with the actual sensation of hearing, including phons and sones, which are based on the equal loudness curves. Detailed description of these is beyond the scope of this thesis, and therefore a single number representing both sound pressure and sound pressure level is adopted, the A-weighted sound pressure level. A-weighting is a frequency weighting defined in the standard IEC 61672-1. It is roughly the invert of the equal loudness curves, and therefore it mimicks the frequency sensitivity of human hearing. When sound pressure level is weighted using the A-weighting scale the obtained A-weighted sound pressure level, denoted as $L_A$, correlates quite well with subjective ratings of sound levels. A curve of the A-weighting in third-octave bands is presented in Fig 3. It can be seen that attenuation increases when frequency lowers and at the low end of human hearing the attenuation is more than 40 dB. Also there is emphasis on the 1,000 – 5,000 Hz range, which corresponds to the resonance frequency of the ear canal.

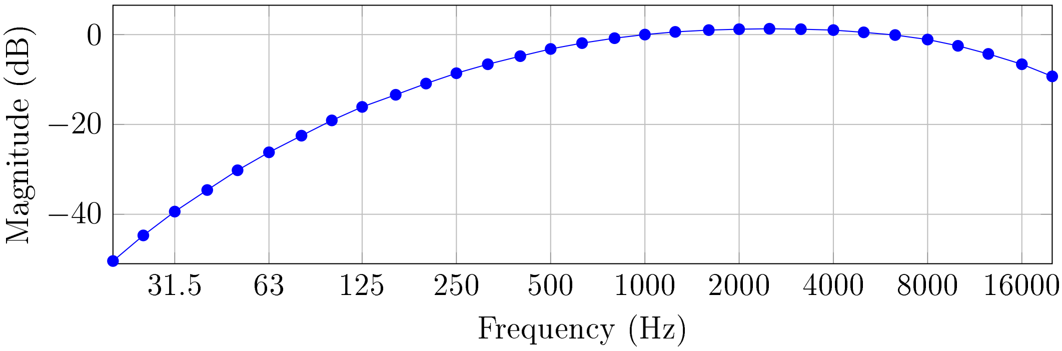


Figure 3: A-weighting curve depicted in one-third octave bands.

## 2.3 Resonance

We have learned that sound consists of pressure fluctuations in air, which implies that two sound waves having the same frequency content colliding together produces either constructive or destructive interference. Depending on if the waves have a pressure maximum or minimum at the point of meeting they will either amplify or attenuate each other. This is the basis of acoustic resonance. Next, basic theory of resonance is given, starting from mechanical resonance and analogies to aocustic resonance. Finally, applications based on the resonance phenomenon that are used in acoustics are presented.

### 2.3.1 Mechanical Resonance

In mechanics, a spring-mass system is a construction of a spring with a given spring constant $k$ and mass $m$ connected to each other as depicted in Fig. 4. If no external force is applied to the system, the mass will remain still since the tension of the spring cancels out the gravity. When an external downwards pulling force is applied to the mass, the spring starts to stretch, i.e. it resists the movement and tries to restore the system to its equilibrium position. If the displacement is noted with y-coordinate, the force applied by the spring when the mass is moved from its equilibrium can be stated as $F = -ky$. If the force applied by the spring is proportional to the displacement, it is called simple harmonic motion. This means that the period does not depend on the amplitude, i.e. the time in which the mass moves from one extreme to the other is constant (Rossing et al., 2002). This system has a nominal frequency which is determined by the spring constant $k$ and mass $m$ as

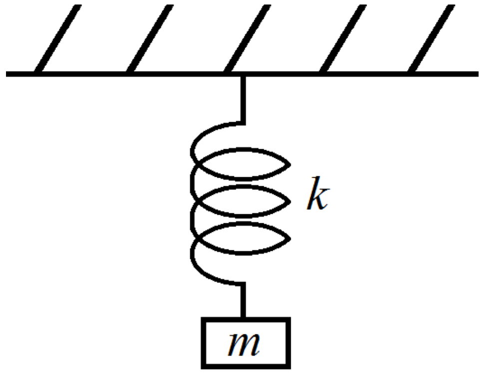


Figure 4: A construction of a spring-mass system with mass $m$ and spring constant $k$.

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}}. \tag{9}$$

The same principle presented in the previous paragraph can be applied to the motion of air, also. The analogy is that the air usually acts as the spring and mass can either be an air column or a solid such as a plywood slab. This requires a certain type of construction for these conditions to meet. An illustration of such construction where the mass is an air column or a plywood slab is presented in Fig. 5. The image on the left of Fig. 5 is known as the Helmholtz resonator, named after the person who used this type of resonators in his acoustical studies, Hermann von Helmholtz. The Helmholtz resonator is discussed in Section 2.3.2. The resonance frequencies for these systems can be calculated, but first the general equation for a cavity covered with a sheet is presented, from which the special cases can be derived. This is defined as

$$f = \frac{c}{2\pi}\sqrt{\frac{\rho}{md}} \tag{10}$$

where $\rho$ is the density of air, $c$ is the speed of sound in air, $m$ is the acoustic mass per unit area and $d$ is the depth of the air cavity. The covering sheet can be perforated to form a perforated panel resonator based on the Helmholtz resonator, or it can be flexible to form a membrane absorber. Equation 10 assumes that the wavelength is much longer than the cavity size (Cox and D'Antonio, 2004). When the wavelength starts to approach the dimensions of the cavity, standing waves are formed in the cavity. Standing waves are discussed in Section 2.3.6

The resonance means that when the system is excited with the resonance frequency, maximum amplitude and energy transfer is obtained. The maximum for the particle velocity of the sound wave is found at the pipe. Depending on how wide the resonance peak is, a Q-factor (Quality factor) can be determined for the resonator. Q-factor describes the relation between the highest peak of resonance and the level where the peak has decreased to 71 % of its maximum. A high value for Q indicates a sharp and high peak and low Q-factor a broader but lower peak (Rossing et al., 2002). Next, resonances in an enclosed space is introduced, followed by presenting the applications of resonance in Helmholtz resonators and panel resonators.

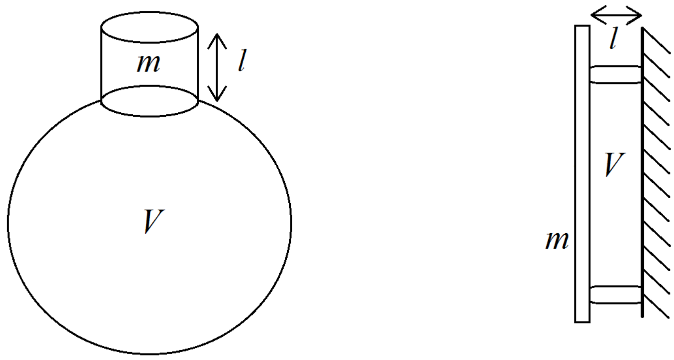


Figure 5: Left is a Helmholtz resonator with a spherical air volume and a pipe connected to it. Right is a panel resonator installed on a wall.

### 2.3.2 Helmholtz Resonator

A Helmholtz resonator is a device with a closed air space and an open pipe connected to it. Together they form an acoustical spring-mass-system where the air in the enclosed space acts as the spring and the air in the pipe as the mass. A nominal frequency for this system can be calculated depending on the shape of the air pipe. Two most common types are introduced here, which include square and circular pipes. The nominal frequency for a square pipe is

$$f = \frac{c}{2\pi}\sqrt{\frac{a}{V(l+0.9d)}} \tag{11}$$

where $a$ is the area of the cross-section of the pipe, $l$ is length of the vibrating air column, $V$ is volume of the air space and $d$ is the square opening's edge length. For the circular opening the equation is

$$f = \frac{c}{2\pi}\sqrt{\frac{a}{V(l+1.6r)}} \tag{12}$$

where $r$ is the radius of the opening (Everest and Pohlmann, 2009). The length of the air column is not straightforwardly the length of the pipe, as can be seen from the last terms of Equations 11 and 12. An appropriate end correction is added to the physical length of the pipe. The end correction applies for the radiation impedance which occurs in the orifice. Detailed explanation of the end correction is not given here, but interested reader is encouraged to read Cox and D'Antonio (2004, p. 166-168), for a more thorough investigation of the subject.

When a sound wave having frequency content in the area of the nominal frequency of the resonator enters the pipe, the air starts to vibrate at its nominal frequency. A common example of this is a glass bottle with some liquid in it. When blown into the neck of the bottle, a distinct note is created. This principle can be used in the design of sound absorbers. As was stated in Section 2.1, the sound pressure level is related to the particle velocity of the air molecules. Reducing the particle velocity of the air molecules results in a reduction in sound pressure level. As can be seen from Eq. 2, the particle velocity is reduced when the density of the medium is increased, but the medium should not be too dense for the sound wave to simply reflect back from the surface. This can be obtained with a porous material, such as mineral wool. Now that the resonant frequency for the resonator can be calculated, inserting absorptive material into the place where particle velocity is largest, i.e. into the pipe where the resonance takes place, maximum absorption is obtained at desired frequency. Care must be taken, however, because adding too much absorption may completely prevent air flow and thus the resonance won't occur. Porous absorber in the air cavity also changes the Q-factor, making the resonance peak slightly lower but broader. The absorbent material is also effective at some distance away from the orifice. A rule of thumb is the length equal to the radius of the orifice. This is due to the consideration that when the sound wave is pushed through a small hole, the particle velocity increases, and when it exits the hole it returns to a free-field circumstances in a gradual manner, and therefore the particle velocity is higher also at some distance away from the orifice. (Cox and D'Antonio, 2004)

The resonator may also have a reverberation time, which can be heard if it is longer than the nominal reverberation time of the space where the resonator is in. Luckily, however, in order for the resonator to have a long reverberation time, it has to have a large Q-factor. Most resonator systems to be applied in practice have low Q-factors, which is more practical because broader absorption is usually preferred instead of a very sharp absorption at one frequency. (Everest and Pohlmann, 2009; Ingard, 1953)

An individual Helmholtz resonator would not be very practical to use in a large space, because the absorption area of a single resonator is quite small. Instead, by perforating a membrane absorber, an array of Helmholtz resonators can be obtained. These membrane absorbers and perforated panel -type resonators are considered next.

### 2.3.3 Membrane Absorbers

Panel, membrane or diaphragm absorbers function in a similar way as the Helmholtz resonators. A panel installed in front of an air cavity forms an acoustic resonator. Here, instead of an air column, the panel acts as a mass that vibrates at its nominal frequency. Absorption happens because the vibrating panel bends and frictional heat losses within the material create damping. When the cavity behind the panel is filled with air the resonance frequency can be estimated as

$$f = \frac{60}{\sqrt{md}} \tag{13}$$

where $m$ is the surface density of panel, in $kg/m^2$ and $d$ is the depth of the air cavity behind the panel. When there are porous material in the cavity, the numerator in Eq. 13 should be changed to 50. The absorption obtained with membrane absorbers is not very effective, since the moving panel also radiates sound. If the air cavity is left empty, maximum absorption coefficient is about 0.3 for a 5 mm plywood with 50 mm air cavity behind it and there is no clear resonance peak visible. Absorption can be increased by introducing porous material into the air cavity. When the air cavity is filled with mineral wool, the resonance is emphasized and the absorption coefficient of the system roughly doubles. (Everest and Pohlmann, 2009, p. 203)

According to Everest and Pohlmann (2009) and Cox and D'Antonio (2004), the Equation 13 is not altogether accurate, because several assumptions about the behaviour of the panel are made. It is assumed that there are no higher order modes in the panel at the frequency bandwidth that is investigated. Also the movement of the panel is assumed to be uniform throughout the panel. However, the panel has to be installed e.g. to a wall, and therefore the attachments, even at the edges, prevent the panel from moving as a one mass. This results in a smaller amount of mass that vibrates creating losses and changing the tuning frequency. To some extent, this can be prevented by attaching the panel with foam that allows the mass to move more freely. The angle of incidence also affects the absorption and the greater the angle the possibility for bending waves to occur increases, which increase the uncertainty of the equation.

Placement of the membrane absorbers should also be considered. Since the absorption effect is based on the vibration of the panel, it is convenient to install the panel where maximum movement for the panel can be obtained. This occurs at places where the sound pressure is at maximum, and in a enclosed space that is at the walls and corners, as will be explained in Section 2.3.6. Therefore by installing the panel there maximum absorption is obtained. While the Eq. 13 presented is not altogether accurate, it gives a good approximation and a starting point for designing and further investigations.

### 2.3.4 Perforated Panel Resonators

A panel made out of e.g. plywood or medium density fiberboard (MDF) with holes drilled equally apart from each other acts as an array of Helmholtz resonators. With this design, a relatively easy application for the Helmholtz resonance phenomenon can be obtained. One wall from a space can easily be covered with a perforated plywood with mineral wool behind it instead of installing individual Helmholtz resonators into the wall. By changing the volume of the air cavity behind the panel, diameter and density of the holes and thickness of the panel the resonant frequency of the system can be tuned. Also the shape of the holes and their layout affects the tuning. With round holes spaced equally apart from each other as depicted in Fig. 6, the resonant frequency can be calculated with equation

$$f = \frac{c}{2\pi}\sqrt{\frac{\pi r^2}{t' D^2 d}} \tag{14}$$

where $r$ is the radius of the holes, $t'$ is the length of the pipe (i.e. panel thickness) with end correction factor, $D$ is the distance of the holes from each other and $d$ is the depth of the air space behind the panel. The area of the holes and total area of the panel can be expresssed as a single number, the perforation percentage. This ratio essentially defines the resonance frequency instead of the size of the holes itself, as could be thought at first. The resonance peak is strongest with a sound wave hitting the panel at a right angle. For oblique angles the resonance is not as strong, specially with lower frequencies, as the interaction between the holes increases. To improve the absorption of diagonal sound waves, the air cavities behind the holes should be separated so that they create individual air volumes and the interaction between the holes is prevented. (Cox and D'Antonio, 2004)

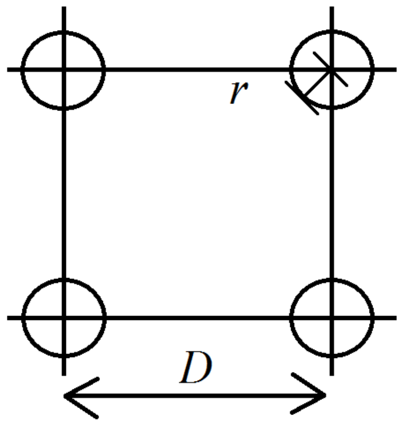


Figure 6: Schematic picture of the perforations in a perforated panel resonator. $D$ is the distance of the perforations from each other and $r$ is the radius of the holes.

It is noted that correct end correction is dependent on the shape of the holes and interactions between resonators that are close to each other. For a single orifice in an infinite plane, the total length is calculated as (panel thickness) + $\delta$ x (hole diameter). The value of $\delta$ depends on the shape of the orifice, as stated earlier. There is some discussion about the correct value of $\delta$. For a single circular hole in an infinite plane, Cox and D'Antonio (2004) uses a value of 0.85 for $\delta$. On the other hand, Everest and Pohlmann (2009) suggests a value of 0.8 to be used.

There are some design limitations to be considered in order for the Eq. 14 to give correct results. The spacing of the holes should be large compared to the diameter of the holes. Also, as stated earlier, the individual holes are not entirely independent from each other at low frequencies and at oblique angles, as the sound waves travel more diagonally. This can be improved by inserting porous absorber to the air cavity, which makes the sound propagation angle more normal to the panel.

From Eq. 14 the effects of different physical measures of the perforated panel on the absorption can be observed. First, as the perforation percentage increases, the resonance peak moves to higher frequencies also. This is intuitive since if the perforation percentage would rise to 100 %, it would mean that there are no perforations at all and the porous material used for absorption would be completely visible. Consequently the porous material would account for the whole absorption, which is considered in Section 3.1. Second, the thickness of the perforated panel, and therefore the length of the air columns in the perforations, has also straightforward effect on the resonance frequency. If the total thickness of the resonator system is kept constant and only the panel thickness is varied, the resonance peak shifts to lower frequencies as the panel thickness is increased. Third, the cavity size behind the panel is connected to the resonance frequency so that when the cavity size increases, resonance frequency moves towards low frequencies.

### 2.3.5 Microperforated Panel Absorbers

Microperforated panel (MPP) absorbers are similar to Helmholtz resonators, but the air cavities in the panel are much smaller than in traditional Helmholtz devices, namely under 1 mm in diameter. When the holes are made sub-millimeter in diameter, viscous losses are introduced to the sound field and therefore there is no need to have a porous material behind the perforated sheet to gain absorption. Microperforated absorbers were first developed in the 1960's, but the theory governing the phenomenon was developed later. The exact theory and design equations were published in 1998 by Maa (1998). Theory of the design parameters for microperforated absorbers are presented here. It is found out that microperforated absorbers have simple and predictable characteristics and that they can be made of various materials, even from plastic which creates a possibility to make an absorber that is transparent. This type of absorber is discussed at the end of this section. Problems in producing microperforated absorbers arises from manufacturing, since very small holes are not trivial to drill.

Three coefficients essentially define the MPP absorber, when the air cavity behind the panel is rectangular. These coefficients are relative acoustic resistance $r$, resonance frequency $f_0$ and perforate constant $k$ of the perforations. This can be derived from the sound absorption equation of the microperforated absorber for a soundwave colliding at a normal angle, which is

$$\alpha = \frac{4r}{(1+r)^2 + (\omega m - \cot(\omega D/c))^2} \tag{15}$$

where $\omega$ is the angular frequency and $D$ is the cavity depth behind the microperforated absorber panel. In the Eq. 15, the last term in the denominator gives the resonance frequency when assigned to zero, and that is also when the absorption reaches its maximum value, as can be expected. With a little mathematics we arrive at the form where the equation for resonance frequency gives the ratio of cavity depth and wavelength of the resonance frequency of $\alpha$ as

$$\frac{D}{\lambda} = \frac{1}{2\pi}\cot^{-1}\omega_0 m. \tag{16}$$

The relative acoustic resistance $r$ is given by

$$r = \frac{32\eta t}{\sigma\rho_0 c d^2}\left(\left[1+\frac{k^2}{32}\right]^{1/2} + \frac{\sqrt{2}}{32}k\frac{d}{t}\right) \tag{17}$$

where $t$ is the depth of the perforated hole, $\rho_0$ the density of air, $\eta$ the viscosity coefficient of air, $d$ the diameter of the holes and $k$ is the perforate constant, which is given by

$$k = r_0\sqrt{\rho_0\omega/\eta}. \tag{18}$$

The second term in the Equation 17 is noted as the resistance coefficient $k_r$, which can be plotted as a function of $k$, as depicted in Fig. 7. In the plot the ratio $d/t$ is assumed as one. From the figure it is noted that $k_r$ is nearly constant for values of $k$ under 1, and it starts to increase dramatically after $k$ is larger. From the Equations 15, 16 and 17 it is seen that the values of $r$, $k$ and $f_0$ determine the characteristics and performation of the microperforated absorber completely. (Maa, 1975)

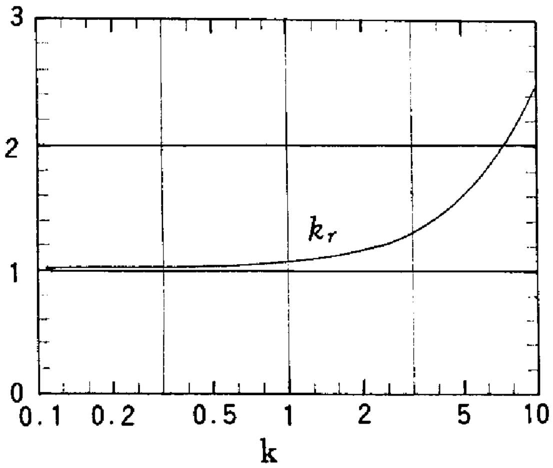


Figure 7: Resistance coefficient $k_r$ plotted as a function of the perforate constant. Note: The scale of $k$ is logarithmic. Picture adopted from Maa (1998).

The absorption bandwidth with the MPP design is quite narrow. An easy way to broaden the effective absorption bandwith was presented by Wang and Cheng (2010) where the backing cavity was made irregular in shape. With this design the absorption after the first resonance peak was found to be significantly higher than with traditional MPP absorber where the cavity was rectangular. The reason for the improved absorption was stated to be that more modes in the cavity is coupled to the microperforations in the panel, thus creating more resonance peaks at higher frequencies and lower dips between the peaks.

Since there is no need for the absorptive material, microperforated absorbers can be made from transparent material and thus make the absorber nearly invisible. The size of the cavities in terms of manufacturing restricts the effective frequency area to low and mid-frequencies. A commercial example of such a system is presented by D'Antonio and Cox (2005).

### 2.3.6 Room Modes

Room modes are a resonance phenomenon which occurs in every enclosed space from living rooms to large concert halls. In the literature there is a number of different names for the same phenomenon: standing waves, natural frequency, room modes or room resonances. In this thesis the name room mode will be used from here on. Depending on the dimensions of the space, the modes can be at very low frequencies when they do not impair acoustics (large concert halls), or at more audible frequencies when they constitute a serious problem, as in studio control rooms or in smaller auditoriums and atriums. The most simple mode is developed between two parallel walls that reflect sound. If the walls are at distance $L$ from each others, the first resonance, and therefore the first mode, occurs at a frequency of $f_1 = c/2L$. This happens because the wavelength of the sound wave is half of the walls' distance, and therefore when the sound wave reflects back from the wall it will be at the opposite phase compared to the original wave. These two sound waves create a room mode with pressure minima and maxima, which is considered next.

In the case of lowest room mode pressure maximum is found at the walls and minimum at the center of the room. In contrast, particle velocity is minimum at the walls and maximum at the center of the room. Resonance occurs also at an integer multiples of $f_1$. These are called axial modes. In a room, modes can develop between all four walls or even between walls, ceiling and floor, as depicted in Fig. 8. These are called tangential and oblique modes, respectively. For a rectangular room, the room modes can be easily calculated using Eq. 19

$$f = \frac{c}{2}\sqrt{\frac{p^2}{L^2} + \frac{q^2}{W^2} + \frac{r^2}{H^2}} \tag{19}$$

where $p$, $q$, and $r$ are integer numbers and $L$, $W$ and $H$ room length, width and height, respectively. The integers $p$, $q$ and $r$ can be used to label axial, tangential and oblique modes. If two of the integers are zero, the mode is axial, e.g. (0, 2, 0), if one is zero, e.g. (2, 1, 0), the mode is tangential, and if none of them are zero the mode is oblique, e.g. (1, 2, 3). Furthermore, the integer represents the frequency multiple of the lowest mode. (Everest and Pohlmann, 2009)

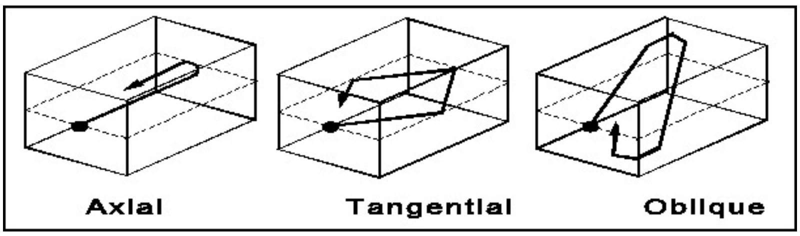


Figure 8: Forming of axial, tangential and oblique room modes. Picture from www.roommodes.com

Since the room modes are governed by the room dimensions, choosing the dimension properly some of the room mode problems can be avoided. A cubical room has same length for all three dimensions, thus creating one very loud room mode as three opposite wall surfaces have the resonance at the same frequency. By designing the space properly, the room modes can be distributed evenly in frequency, which smoothens the resonance peaks. A number of suggestions of room proportions for a rectangular rooms have been given, which should produce evenly distributed room modes. Every proportions that are multiples of each others are problematic, since the dimensions produce amplification to each others' modes. With a dimension ratio of $1:2:3$ the modes are spread out more evenly. Using non-integer multiples even better mode distribution can be obtained, e.g. using the ratio $5:3:2$ or the golden ratio $1.618:1:0.618$. Oblique walls can also be used to spread the axial modes, but this way tangential and oblique modes becomes more hard to predict, which can result in problems that cannot be predicted. (Rossing et al., 2002)

# 3 Sound Field Treatment

In this section is presented the basis of how a sound field in an enclosed space can be treated.

## 3.1 Absorption and Insulation

As stated in Section 2.1, sound is energy that consists of vibrating air molecules. In a free field conditions, e.g. outdoors, a sound wave from a unidirectional point source is expanding to the growing sphere surface, thus decreasing the sound intensity as the distance grows. Since, according to the first law of termodynamics, energy cannot be destroyed, reducing sound energy in an enclosed space where free field conditions do not apply and sound is reflected back and forth means that the sound energy has to be transformed into a different form, e.g. to heat. The property of a material to convert sound energy into heat is called absorption. For every material a quality called absorption coefficient can be determined, which defines the ratio of how much of the sound wave is transformed into heat. There are many ways to measure it, as listed by Cox and D'Antonio (2004). Absorption is also highly dependend on the incident angle and frequency of the sound wave. (Everest and Pohlmann, 2009)

### 3.1.1 Absorption Coefficient

The absorption coefficient is defined as the ratio of sound intensity hitting the material and intensity absorbed by the surface. A material's absorption coefficient depends on the incident angle of sound wave, and the angle should be defined when an absorption coefficients for a material are presented. Another way to employ the incidence angle is to define the absorption for a diffuse sound field, which means that the absorption is averaged over all incident angles of the hemisphere. In this thesis, $\alpha$ is adopted to refer to the absorption coefficient defined in a diffuse sound field. From that the absorption coefficient is defined as

$$\alpha = \frac{I_i - I_r}{I_i} \tag{20}$$

where $I_i$ is the intensity hitting the material and $I_r$ the intensity reflected back from the material.

The absorption coefficient is frequency dependent and it is usually reported within octave bands ranging 125 – 4000 Hz. Based on this theory, absorption coefficient can have values from 0 to 1, where 0 means that no sound energy is absorbed and 1 meaning that all sound energy is absorbed. There are also measurements of absorption coefficients higher than 1. This is usually a product from diffraction and edge effects that happens at the edges of the material. Absorption coefficients higher than one can also be measured if other measurement techniques are used, e.g. the reverberation chamber method. Using this and Sabine's Equation for reverberation time, which is defined later, absorption coefficients higher than one are possible. This is discussed further in Section 3.2. From the absorption coefficient total absorption area can be calculated for an object or surface, which is simply the product of the area of the object and its absorption coefficient. Absorption coefficient itself is unitless. (Hongisto, 2012; Cox and D'Antonio, 2004)

### 3.1.2 Porous Materials

While every material has an absorption coefficient, one material category is the most widely used to introduce absorption into a space. These are porous materials, especially mineral wool, because they are proven to be very effective in absorbing sound energy. Their ability to convert sound energy into heat is based on two fundamental properties, the material's flow resistivity $\sigma$ and porosity $\varepsilon$. Porosity is the ratio of the air volume inside the absorbent to the overall volume of the absorber. For most types of mineral wool, the porosity is close to 1, and therefore flow resistivity is the more dominant property that determines the absorption of a porous material. Flow resistivity is, as the name states, the materials property of how easily air can flow through the material and how much resistance it encounters on its way. This is defined as

$$\sigma = \frac{\Delta P}{Ud} \tag{21}$$

where $U$ is flow velocity of the air, $d$ is the thickness of the material and $\Delta P$ is the pressure drop from the side where the air flows to the other side.

From the porosity and flow resistivity the absorption quality of a material can be estimated, but the actual absorption depends, in addition to the incident angle and frequency, also on the physical dimensions of the absorptive material. Thickness of the material has a quite straightforward effect on the absorption properties of a material. When the thickness of the material is rougly one-fourth of the wavelength, full absorption occurs. When approaching the one-fourth wavelength the absorption coefficient starts to rise gradually. In an enclosed space, good absorption can be achieved with thinner absorptive materials also, if it is installed away from walls. This happens because at the walls the pressure is at maximum and particle velocity is at minimum and particle velocity starts to increase when going further away from the walls and thus the absorptive material has greater effect on the absorption. In conclusion, if low frequency absorption is desired, porous material has to be either installed about quarter-wavelength away from a wall or the material has to about quarter-wavelength in thickness. (Cox and D'Antonio, 2004; Hongisto, 2012)

### 3.1.3 Sound Insulation

In the previous paragraph it is assumed that all sound energy is either absorbed or reflected back from the surface. This is true when the absorptive material is installed on a solid surface that does not absorb or let through any sound energy. Sound energy can also penetrate the absorptive material, which usually happens if it is installed freely to a space with no reflective surfaces near. In this case the sound energy is neither reflected back nor absorbed by the material and this sound energy does not contribute to the absorption coefficient of the material. It defines the sound insulation of the material, and this should not be mixed with the absorption coefficient. Sound insulation expresses the ratio of incident and penetrated sound energy, and it is noted in decibels. Sound insulation is calculated with

$$R = 10\,\log_{10}\frac{1}{\tau} = 10\,\log_{10}\frac{I_i}{I_t} \tag{22}$$

where $\tau$ is the penetration ratio and $I_t$ is the penetrated sound intensity.

Likewise the absorption coefficient, the sound insulation is also strongly dependent on the frequency. For normal materials the sound insulation is between 0 – 70 dB. An illustration of the differences in the physical phenomena of absorption and sound insulation is presented in Fig. 9. In Fig. 9 A a porous material is installed on top of a hard surface. When a sound wave strikes it, absorption occurs in the porous material and, to some extent, also in the hard material. Sound wave that is not absorbed or does not penetrate the wall is reflected back. The bolded arrow is the incident sound wave, thinner arrow is the reflected sound wave and dashed line arrow represents the penetrated sound wave. This configuration gives both good absorption and sound insulation, and therefore the penetrated intensity $I_t$ is much smaller than the incident intensity $I_i$. In Fig. 9 B only porous material is presented. The same absorption occurs as in A, but since there are no hard wall behind the absorbent, sound insulation is poor and the incident intensity $I_i$ is only a bit smaller than the penetrated intensity $I_t$. (Hongisto, 2012)

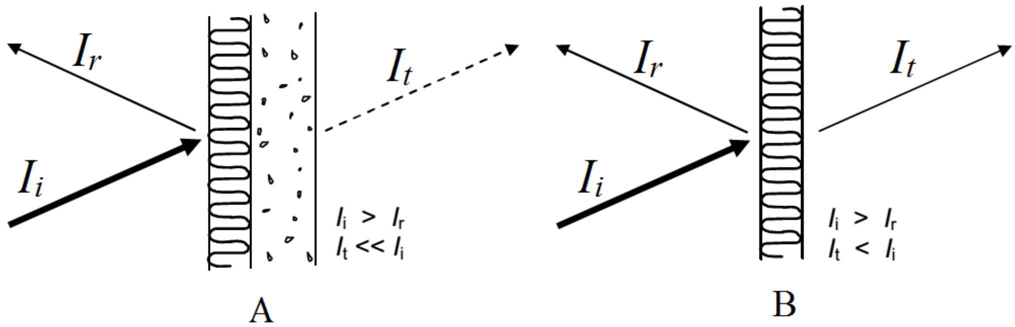


Figure 9: Sound insulation (A) and absorption (B). Figure adopted from Hongisto (2012).

## 3.2 Reverberation

Reverberation, resonance or echo may be mixed with each other, so it is convenient to define these phenomena in order to avoid confusion. Echo is a single reflection of a sound from a hard surface that can be heard separately from the surrounding sound field. Reverberation is used to describe the overall sound that consists of multiple echoes coming from all directions. Resonance was already defined in Section 2.3. While this nomenclature is quite straightforward, there are some mixing and confusion in the literature and therefore it is made clear here and used throughout this thesis.

### 3.2.1 Reverberation Time

In an enclosed space sound is reflected, absorbed and penetrated through the surfaces. All this motion of sound combined together produces reverberation field in the space. The time that elapses when a sound source in a space is muted and the sound is attenuated to one-millionth of its original value, which corresponds to 60 dB, is called reverberation time ($T_{60}$). It can be determined either with measurements or by calculation using the absorption area and the volume of the space. Measurements are usually done using strong impulse or by first playing back pink noise at a high level and then suddenly cutting it off. In both cases the sound pressure level in space and elapsed time is measured after the source is muted. However, for practical reasons it is hard to obtain over 60 dB signal-to-noise ratio, and therefore the measurements are usually done from -5 to -25 dB or -5 to -35 dB range and then multiplied by three or two, respectively, in order to obtain the full 60 dB decay. Reverberation times obtained with these methods are noted using $T_{20}$ or $T_{30}$.

### 3.2.2 Sabine and Eyring Equations

In computation of reverberation time it is assumed that the sound field is diffuse, that is, the sound pressure level in every point in space is constant and the intensity is zero. In practice these kind of circumstances never exists, but the assumption is close enough for the calculation. The first theoretical suggestion for calculating reverberation time was introduced by W. C. Sabine, and the equation is named after him as the Sabine equation

$$T = 0.16\frac{V}{A} \tag{23}$$

where $V$ is the volume of the room and $A$ is the total absorption area in the room (Sabine, 1922). The absorption area for $n$ different surfaces for a given frequency band is calculated by

$$A = \sum_{i=1}^{n} \alpha_i S_i \tag{24}$$

where $S_i$ is the $i$th surface that has an absorption coefficient of $\alpha_i$.

The Equations 23 and 24 are adequate for calculating the reverberation time assuming the space is cubic in shape, the amount of absorption is low and it is evenly distributed within the space. This can be interpreted as Beranek (2006) says, 'as a sound wave travels around a room it encounters surfaces "one after another"'.

From the Eq. 23 it can be seen that in order for the reverberation time to reach zero, the total absorption area has to be infinite. This, in consequence, implies that the absorption coefficient should be infinite because the physical area cannot be infinite. As was stated in Section 3.1, absorption coefficients have values from 0–1, and this seems to lead to a conflict. While absorption coefficients over one seem physically impossible, Beranek (2006) has stated that this is not the case, but rather a quality of the Sabine's Equation and one must accept absorption coefficients higher than one if they are using the equation. However, problems arise when measured absorption coefficients assuming the range from 0 (no absorption) to 1 (all energy is absorberd) are used with Sabine Equation. This can be confusing and also to some point unintuitive, so other solutions to avoid conflicts are sought after.

Since the pioneering work of Sabine, numerous improvements to the equation in order to overcome the limitations of Sabine's equation has been proposed. The most widely used of these is the Eyring's equation, which is defined as

$$T = \frac{-0.16V}{S\,\ln(1-\bar{\alpha})} \tag{25}$$

where $\bar{\alpha}$ is the average absorption ratio and $S$ the total area. Average absorption ratio is defined as

$$\bar{\alpha} = \sum_k \alpha_k S_k / S \tag{26}$$

where $\alpha_k$ is absorption coefficient and $S_k$ is the area of surface $k$ (Hongisto, 2012). The Eyring equation alleviates the need to use higher than 1 absorption coefficients. Eyring equation is also designed for rooms that have more absorption in them, because the Sabine equation's need for higher than 1 absorption coefficients may not work correctly when there are high amount of absorptive material (Passero and Zannin, 2010).

It is noted in many references (Barron, 2009; Beranek, 2006) that the reverberation time obtained with Sabine or Eyring equations is usually higher than the actual reverberation time obtained with measurements. Beranek (2006) lists a few studies where the differences between modeled and measured reverberation times are 0.3-0.4 seconds. Several reasons are suggested for this. In order to use the Sabine Equation, the absorption coefficients for the materials should be determined at a similar space in where the equation is to be used. In consequence, absorption coefficient higher than one should be accepted. Also the limitations listed when the Sabine equation was introduced contribute to the elevated reverberation time. Eyring equation is linked to the Sabine equation by the logarithm in the denominator, and therefore absorption coefficients applicable to the Eyring equation can be derived from the Sabine versions. This way both equations can be used to calculate the reverberation time of the space for a more comprehensive estimate of the real situation.

## 3.3 Diffusion

This section introduces the different phenomena of sound waves reflecting from surfaces.

### 3.3.1 Specular and Diffuse Reflections

Sound wave with a wavelength smaller than the dimensions of a smooth wall reflects from it following Snell's law, where the reflection angle is equal to the incidence angle. This is mathematically expressed as

$$\frac{sin(\theta_1)}{sin(\theta_2)} = \frac{c_2}{c_1} \tag{27}$$

where $\theta_1$ is the incident angle, $\theta_2$ the angle of reflection, $c_1$ and $c_2$ are velocities of the wave in the material where the wave is coming from and the material it is reflected from, respectively. This reflection is called the specular reflection and in an enclosed space it produces loud reflections to one direction which can be heard as separate echoes and not as part of the reverberation. Strong reflections may also create an illusion of the sound source to be at the direction of the reflective surface instead of the direction of the actual source. In the case where two reflective surfaces are parallel to each other a flutter echo may also form. A flutter echo is a sound wave reflecting back and forth from two reflective surfaces, creating a distinct sound that is heard longer than the nominal reverberation time of the space.

Instead of one loud reflection or flutter echoes it is desirable to have reflections with even loudness from many different directions so that the sound field is more constant in every point of the space and the sense of direction for the sound source is undisturbed. This is true because the direction of a sound source is mainly governed by the first sound arriving at the ear. This is called the precedence effect (Karjalainen, 2009). Therefore having diffusion in a space is often a desirable quality. This holds for nearly every space where humans spend time, not only for large concert halls. This can be achieved with a proper amount of diffusive materials in a space that distributes the sound energy more evenly. This also decreases large spatial deviations in the sound pressure levels, which makes a space acoustically more comfortable to spend time in. (Dalenbäck, 1994; Cox and D'Antonio, 2004)

When a sound wave reflects to other direction than what the specular reflection assumes, the surface is said to be diffusive. This diffusion of a diffusive surface can be seen as its property to change the direction in which the energy is reflected, i.e. away from the specular direction. Here is a distinction between absorption and diffusion. While absorption removes total sound energy from the sound wave, diffusion should only distribute the same amount of energy into different directions, as depicted in Fig. 10. (Dalenbäck, 1994)

Diffusion and scattering are properties of a surface, and there exists measures for both, the diffusion and scattering coefficients. They are often used as meaning the same thing, but there is a fine distinction between them. As Cox and D'Antonio (2004) states, diffusion coefficient, which measurement is defined in the standard ISO 17497-2, is a measure for the uniformity of the reflected sound field from a surface. In contrast, scattering coefficient is defined as the sound energy not reflected by following Snell's law relative to the total reflected sound energy. This nomenclature is also used in this thesis. The differences in their definitions is why they are used for different design problems, although they seemingly state the same thing.

Diffusion coefficient is defined by Eq. 28

$$d_\psi = \frac{\left(\sum_{i=1}^{n} 10^{L_i/10}\right) - \sum_{i=1}^{n} (10^{L_i/10})^2}{(n-1)\sum_{i=1}^{n} (10^{L_i/10})^2} \tag{28}$$

where $L_i$ is the sound pressure level in dB, $n$ is the number of receivers in the

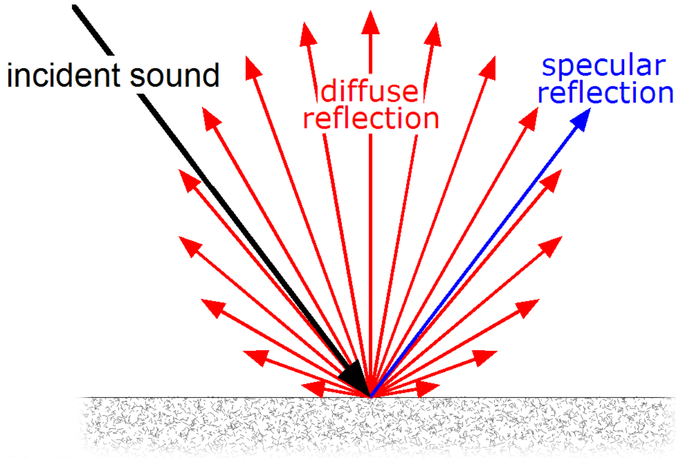


Figure 10: Illustration of a diffusive surface that distributes the incident sound wave into multiple directions and the specular direction is only a small part of the reflection. Picture adopted from Wikimedia Commons.

polar response and $\psi$ is the incidence angle. This equation, which is based on autocorrelation function, gives high values when the investigated sound pressure levels are high in all directions of the angle $\psi$, and vice versa when only one angle of $\psi$ has high value. In other words when the measured surface distributes sound energy to all directions it has a high diffusion coefficient. This way it is reasonable to compare different diffusive structures to each other with comparing their diffusion coefficients. This is also beneficial for someone designing and developing diffusers. (Cox and D'Antonio, 2004)

On the other hand, scattering coefficient holds the information for which part of the sound wave is reflected in a specular manner and which is scattered. Mathematically the scattering coefficient is defined as

$$s = 1 - \frac{E_{spec}}{E_{total}} \tag{29}$$

where $E_{spec}$ is the sound energy that is reflected in the specular direction and $E_{total}$ is the total incident sound energy that arrives at the surface. This information is useful for an acoustician using a computer model for the calculation of indoor acoustics. By defining scattering coefficients for a surface the software is able to calculate sound field in a more realistic way instead of calculating only specular reflections, which improves the accuracy of the modeling. This is discussed further in Section 4.3.1.

### 3.3.2 Diffusers

Diffusion can be achieved with dispersing the sound waves in to the space using specific structures designed to scatter sound. This type of construction is called a diffuser. Diffusers are uneven surfaces that scatter the incident sound waves to different directions instead of reflecting it only to one direction. In addition to their diffusive characteristics, Dalenbäck (1994) gives a general statement about diffusers' basic frequency response, stating that when compared to a flat surface, diffuser acts as a low-pass filter. This is based on the physical quality that in order to have diffusion in the low-frequency range, the dimensions of the diffuser needs to be reasonably large.

Dispersion of a sound wave can be achieved in time or in space. Spatial dispersion can be used to achieve uniform sound field over the desired space. If in a space there is coloration of the sound, temporal and spatial dispersion is needed in order to take care of it. It also happens, that a diffuser that produces only spatial dispersion, e.g. a semi-cylinder, introduces strong comb-filtering, which makes the sound unpleasant. Diffusers can be categorized into 1- or 2-dimensional diffusers, depending on how many directions they provide diffusion. (Cox, 2005)

One of the most popular diffusers used are the quadratic residue diffusers (QRD), also known as Schroeder diffusers, named after their inventor, Manfred Schroeder. They are diffusers that have wells with depths based on quadratic residue sequences. The most important quality of a QRD is that the energy scattered by it is constant to all directions. They should also not provide unwanted absorption, if they are made of non-absorptive material, even though their operation is based on resonance. (Rossing et al., 2002) Quadratic residue sequences are defined by

$$s_n = n^2 \bmod N \tag{30}$$

where $n^2$ is taken as least non-negative remainder modulo $N$ and $N$ is a prime number. E.g. for $N = 5$ the sequence is $s_n = \{0, 1, 4, 4, 1\}$. An illustration of a QR-diffuser with $N = 5$ is depicted in Fig. 11. In order for the QRD to work correctly, plane wave propagation must occur in the wells. Therefore each well has to be separated from each other with thin but rigid walls. An upper limit for the diffusion to follow the design theory is found from

$$w = \lambda_{min}/2 \tag{31}$$

where $\lambda_{min}$ is the minimum wavelength before modes start to develop in the wells and $w$ is the well width. Above this frequency diffusion happens also, but it becomes more complex to model.

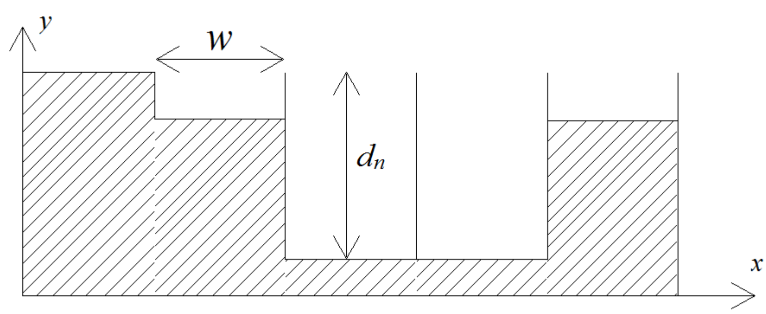


Figure 11: A cross-section of a quadratic residue diffuser with $N = 5$.

The well depths are defined by the desired operation frequency and its wavelength, which is the lower frequency limit of the diffuser. The limit does not implicitly represent the lowest frequency where the diffuser provides more diffusion compared to a smooth surface. Instead it expresses the lowest frequency where the energy scattered by the diffuser can have even diffraction lobes. Since the diffuser has physical dimension, it is more convenient to express the design frequency in terms of its wavelength. Thus the depth of $n$th well of the diffuser is defined as

$$d_n = \frac{s_n \lambda_0}{2N} \tag{32}$$

where $\lambda_0$ is the design wavelength. Eq. 32 states that the depth of the wells vary between a range from 0 to $\lambda_0/2$. (Cox and D'Antonio, 2004)

# 4 Literature Review and Design Tools

In this section the Finnish building regulations and guidelines concerning sound insulation and acoustic conditions are presented. Also a brief literature review is made concerning similar projects done in the past. Finally, the software used in the calculations is introduced.

## 4.1 Finnish Building Regulations and Recommendations

In Finland there are very few regulations and guides for the acoustic conditions concerning public spaces. The Finnish building regulations collection RakMK C1-1998 gives regulations for structural sound proofing, noise control from Heating, Ventilation and Air Conditioning (HVAC) and acoustical conditions, namely the reverberation time. For the most parts these regulations concern only habitable buildings, and no regulations for public spaces are given. Also the regulations are mostly for sound insulation, and the only regulation concerning acoustic conditions is the reverberation time limit of 1.3 seconds for hallways in apartment buildings. However, the C1-1998 also states that a space has to be acoustically suitable for its designed purpose of use, so that while no strict regulations in numbers are given, it does not mean that the acoustics of the space can be left to no attention at all. All other limiting numbers are only given as guidelines. E.g. in classrooms the recommended reverberation time is between 0.6–0.9 seconds in order to ensure sufficient speech intelligibility. In dining rooms the recommendation is 1.0–1.3 seconds.

The newer standard SFS 5907 supplements the older RakMK C1-1998 regulations. The standard is not a regulation but a guideline for constructors, property owners and consultants in designing the acoustical conditions based on the purpose of use of a building. The standard defines four different acoustic classes for buildings, A, B, C and D. A has the most strict requirements and C is the minimum level for new buildings. The class C is equivalent to the regulations given in C1-1998. D class is only used for old buildings. The classifications can be made for single rooms or for entire buildings. In order to assign an acoustic class for a room or a building, the building materials have to meet the limits of the target class and measurements has to be performed to ensure that the limits are fulfilled. In the case of an entire building it is not feasible to measure every room. Instead the standard requires that 5 % of the rooms and structures has to measured, and always a minimum of two measurements are required, if the number of rooms to be measured based on the 5 % rule is smaller than two. In order for a building to achieve the target class, all of the measurements have to fulfill the corresponding requirements.

A summary of the regulations and guides concerning reverberation times for different acoustic classes are presented in Table 1. From the table it can be seen that most of the reverberation time limits are between 1.0 – 1.3 seconds, except the school gym which has a minimum requirement of 1.90 seconds. This is understandable since gym classes usually have large, smooth surfaces and much larger volume than other spaces listed in the standard SFS 5907 and hence the initial reverberation time is longer. For this thesis, the suggestions for reverberation time of 1.30 seconds in the office lobbies and 1.90 seconds for gym classes are the closest in terms of volume and use of space. Therefore these are selected as one of the references when designing the food court area.

Table 1: Regulations and guidelines for sound insulation and acoustic conditions in different buildings as presented in RakMK C1-1998 and SFS 5907.

| Space | Coefficient | A | B | C | D |
|---|---|---|---|---|---|
| Hallway | $T_{60}$ | 1.0 | 1.0 | 1.3 | 1.3 |
| Office lobby[1] | $T_{60}$ | 0.80 | 1.00 | 1.30 | 1.30 |
| School gym [2] | $T_{60}$ | 1.30 | 1.30 | 1.90 | - |
| Music class [3] | $T_{60}$ | 0.8–0.9 | 0.8–0.9 | $< 1.0$ | - |
| School hallways | $T_{60}$ | $< 0.9$ | $< 0.9$ | $< 1.3$ | - |
| School lobby | $T_{60}$ | 0.7–0.9 | 0.7–0.9 | 0.9–1.1 | - |
| Between apartments | $R'_w\ (+C_{50-3150})$ | 63 | 58 | 55 | 49 |
| Hallway to apartment | $L'_{n,w}(+C_{l,50-2500})$ | 49 | 53 | 63 | 68 |
| Within apartments | $L_{A,eq,7-22}$ | 25 | 30 | 35 | 35 |

## 4.2 Earlier Studies

There has been a number of previous studies concerning the acoustics of public spaces and the correlation between measured and simulated reverberation times. In this section, a review of a few similar studies is made.

### 4.2.1 Atrium in an Office Complex

Mei and Kang (2012) studied the spatial distribution of reverberation time and sound pressure in a similar atrium space as is concerned in this thesis. The test measurements were conducted in an office complex that had a glass roof with other surface materials being mainly concrete and wood, along with large glass walls. The main atrium consisted of a rectangular hallway with height of 23 meters, which is openly connected to smaller corridors at three sides of the atrium. The volume of the main atrium is around 24,000 m$^3$. In the study, three acoustical attributes that are characteristic for an atrium-type space are recognized. These are large volume, various different shapes and the interaction with other spaces, such as hallways and entrances to shops.

According to the results of the study the sound pressure levels in the ground floor level attenuates when moving away from the sound source. Measurements of the reverberation time revealed that there was a 100 % variation of the reverberation time when both source and receiver was at ground floor. This is depicted in Fig.

[1] Atriums with height over 4 meters require separate designing.
[2] Height over 5 meters
[3] Acoustical design of music classes and auditoriums is recommended.

12, from where it is seen that reverberation time is much higher near the walls, especially if there are two walls close and parallel to each other, which create a room mode. The majority of the measured reverberation times at 500 Hz were between 2–3 seconds, but as the 100 % difference indicates, the reverberation time was not uniform throughout the lobby. These results indicate low diffusion in the space, and thus the acoustical conditions in the space vary with position of the observer. Since the atrium studied is close to the food court area designed in this thesis, these observations about the acoustical characteristics provide useful comparison for the design.

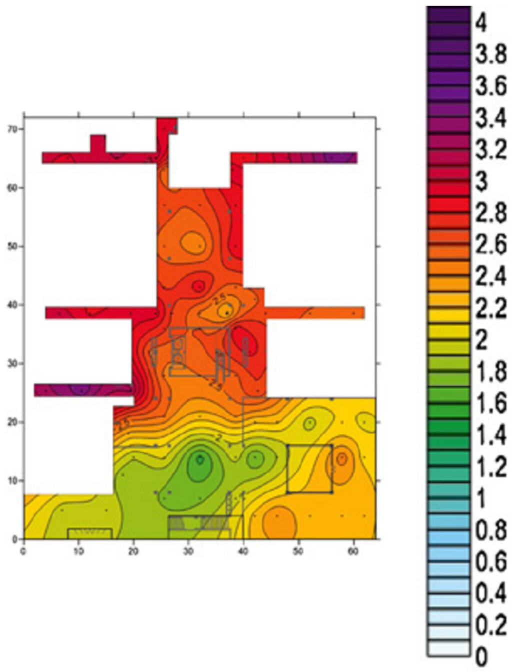


Figure 12: Reverberation time ($T_{20}$) at 500 Hz frequency measured at the ground floor level of an atrium. Figure adopted from Mei and Kang (2012).

### 4.2.2 Acoustic Comfort of Shopping Mall Employees

In a study made in Italy by Crociata et al. (2013), a subjective and objective test was conducted about the acoustical environment of employees in a large shopping mall. The mall was divided into 'silent' and 'noisy' areas which were to be evaluated. Silent areas consisted of pharmacy and electronics sales areas. Noisy areas included bakery products, butcher's desk, the proximity of the registers and warehouse

From the survey conducted for the mall employees the acoustical environment was noted to be satisfactory at silent selling areas for most of the workers. In the silent areas, the source that were reported to be the most annoying was background music and the moving of pallets. At the noisy areas, the objective evaluation did not correlate well with subjecive ratings about the acoustic environment. There was also clear difference in the sound scape at different areas. At the checkouts a steady sound pressure level ($L_{Aeq}$) fluctuating around 70 dB was measured at a 15 minute period. However, at the warehouse the background noise level was much lower, but on the other hand the fluctuation of the sound pressure level was also greater. Highest levels measured were almost 90 dB. While there was higher sound pressure levels, the sound scape was not rated to be uncomfortable. This was due to the fact that most of the noise is generated by the employees themselves, and also because workers are used to noise and are expecting a higher sound levels in these areas. It was also noted that at the registers irregular sound spectrum increased dissatisfaction to the acoustic environment. Also the most annoying sound source

### 4.2.3 Acoustics of an Ecological Building

In a study made by Newsham et al. (2013) the acoustical properties of ecologial buildings were inspected by comparing twelve regular buildings and twelve buildings that were constructed from an ecological point of view. In the study, ecological meant that the building had a LEED (Leadership in Energy and Environmental Design) certification.

Based on the results the ecological buildings were found to be better controlling noise from HVAC systems, but in general the acoustical properties were equally bad or worse compared to the regular buildings. One factor infuencing this is that the LEED does not have credits for acoustic conditions in buildings, but hard ceilings and walls are credited for better air quality, which, in turn, have detrimental effects for the acoustical conditions. From here a conclusion can be drawn that acoustics is not the main priority when designing a buildings, even though the starting point is ecological building. There seems to be even a small conflict since the LEED certificate favors materials that impairs the acoustical conditions.

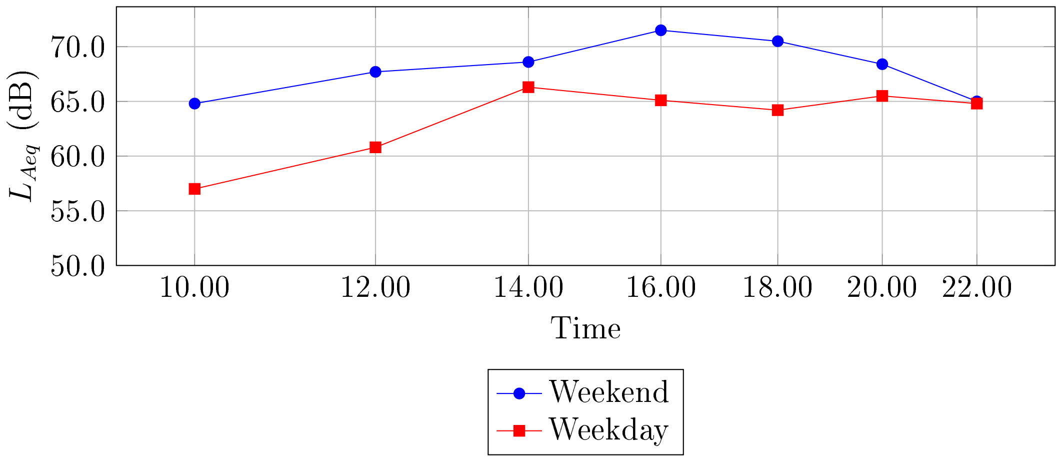


Figure 13: Measured $L_{Aeq}$ levels during weekday and weekend at a CEPA shopping center in Ankara, Turkey. Figure adopted from Döcmeci and Yilmazer (2013).

### 4.2.4 Shopping Center in Ankara

A study conducted in Turkey (Döcmeci and Yilmazer, 2013) concentrated on finding the relationship between objective acoustical measurements and subjective evaluations in a shopping center in Ankara. A food court area open to three floors, with a volume of about 30,000 $m^3$ was researched. It has a glass ceiling, most of the walls being also glass and the floor was granite. No specific attention during the design phase was made to the material choices in order to enhance the acoustics of the space.

Background noise level was measured, being 44 dB, weekday average sound level was $L_{Aeq}$ = 63.5 dB and in weekends $L_{Aeq}$ = 68.3 dB. At weekdays the highest sound levels were during evening hours, and at weekends during the afternoon. A plot of the sound level during weekdays and weekends is presented in Fig. 13. A clear fluctuation in sound pressure levels depending on time of the day and between weekends and weekdays can be observed. Subjective evaluations correlated well with the measured sound levels. Based on a survey conducted on people at the food court area, the area is used mostly to eating, conversation and drinking coffee. In conclusion, too long reverberation time which contributes to the overall noise disturbs the activities that is most often done in a space like this.

### 4.2.5 Simulated Reverberation Times Compared to Measurements

In studies made by Passero and Zannin (2010) and Astolfi et al. (2008), a comparison between simulated reverberation times using ODEON 9.0 and 6.5 ray tracing softwares, calculations with Sabine and Eyring equation and measurements using cut off noise and impulse response methods was made. The rooms that were investigated were classrooms with volumes varying between 160–466 $m^3$. The analysed rooms were furnished, and for the results presented here the rooms were unoccupied.

When comparing the computer simulations to the measured values at 125–4,000 Hz octave bands, the tendency was that the computer simulations gives higher values than the measurements. The average difference was about 10 %. Similar results are found when comparing the Sabine and Eyring reverberation times to the measurements. At higher frequencies this became more pronounced. It is therefore assumed that the reverberation times obtained from the simulations and calculations are somewhat overestimates from the actual situation.

### 4.2.6 Notes From The Previous Studies

From the studies presented here a few key points can be noted. These include the following:

- Large atriums do not have uniform reverberation time
- Fluctuation of the reverberation time in different points of a space may be over 100 %
- Assumptions of the sound scape affects the subjective experience

- Softer sounds in quieter places may be more annoying than louder sounds in a louder environment
- Ecological buildings do not excel in acoustical conditions
- Sound pressure levels ($L_{Aeq}$) in a shopping center may be over 65 dB
- Reverberation times of a typical atrium is generally between 2–3 seconds
- Without acoustical design the acoustic conditions may be poor
- Simulated reverberation times are usually overestimates when compared to measurements

These observations indicate that the food court area presented in the next section will need acoustical treatment in order for the reverberation time to be at acceptable levels.

## 4.3 Design Tools

This section introduces the two main softwares used in the computer simulation, CATT Acoustic v. 9.0 and WinFlag 2.4. The 3D-modeling is done with CATT and calculations of the resonators are done using WinFlag. The 3D-model can be constructed by importing a complete model made with some other 3D drawing software, and then defining properties of the surface materials accordingly. Another possibility is to create the whole space by defining separately every surface in the space, finally making it an enclosed room.

### 4.3.1 CATT Acoustic

CATT Acoustic v. 9.0 is a software for room acoustics simulations, which has been used in creating the acoustical 3D-model for this study. It is based on a ray tracing method. The ray tracing is a method where the soundwave is traced from the source into the space and when it reaches a wall, it is reflected.

The source from which the sound is emitted can be assigned to be omnidirectional or that it has directivity that can be defined in propagation angles at 15 degree spacing. Also the frequency characteristics of the source can be defined in octave bands ranging from 125–16,000 Hz. Also the surfaces can be modeled to have some diffusive characteristics or the reflection to be completely specular. The diffusiveness of the surfaces are given in scattering coefficients for the reasons stated in Section 3.3.

The usual assumption for the surfaces is that the reflections are specular, i.e. the angle of incidence is the same as the angle of reflection. This is accurate for wavelengths that are small compared to the reflective surface. For most surfaces, this is not always the case, at least for the whole frequency band. To know the diffusive properties of surfaces is essential for the acoustic model to give accurate and reliable results for the calculations. As stated earlier in Section 3.1, the calculation of reverberation time using Sabine and Eyring equations assumes absorption material to be distributed evenly to the whole space, which also means that the sound field in the space should be as uniform as possible. Also many spaces have more diffusive surfaces than is feasible to include in detail to a computer model. The diffusion in the room affects the reverberation time in the room (Bork, 2000), and including even a mild diffusiveness to every wall usually leads to a more precise results. (Dalenbäck, 1996)

With the ray tracing method it is easy to employ diffuse reflections to surfaces. If a ray hits a wall that is diffusive, the direction for the reflected ray is determined by randomly generated number. Based on the calculation power and time available, the number of calculated rays reflected from the surfaces can be increased to distribute energy more evenly and to get more accurate results. Diffusion of a surface is given to the software as a scattering coefficient, which tells the percentage of how much of the sound energy is reflected to a non-specular direction. For the complete calculation, absorption of the surfaces has to be accounted for also. This leads to two frequency-dependend coefficients that characterize each surface: the absorption and scattering coefficients. A simple relation between the two coefficient describing the preservation of energy exist as

$$\alpha + \delta(1-\alpha) + (1-\delta)(1-\alpha) = 1 \tag{33}$$

where $\alpha$ is the absorption coefficient and $\delta$ is the scattering coefficient. In Eq. 33 the first term equals as the absorbed energy, the second term defines the diffused energy and last term denotes the energy fraction reflected in the specular direction. This equals as one so that no energy is lost in the simulation, as stated in Section 3.1.

With CATT a number of different measures can be calculated from the space that is modelled, but in this case study the main interest is in the reverberation time, as in the previous studies presented in Section 4.2, it is found to be the most defining quality of the acoustic conditions of such spaces. Both Sabine and Eyring reverberation time equations are used to calculate the reverberation time. As stated in Section 3.1, the Sabine equation gives accurate results if the amount of absorptive material is fairly low and it is uniformly distributed in the space that is cubical in shape. Eyring equation works better if there are a high amount of absorption. The reverberation time is also calculated with CATT Acoustic using the ray tracing method, and all these values are then compared to each other in order to have a thorough evaluation about the reveberation characteristics of the space. All calculations are done with air temperature being 20 °C, relative humidity of 50 % and air density of 1.20 kg/m$^3$.

### 4.3.2 WinFlag

Winflag (2013) is a program used to calculate different type of resonators. It is based on a similar transfer matrix solution that is presented by Cox and D'Antonio (2004, p. 174–178), which calculates the impedances on the surfaces of different material layers. It can calculate perforated panel resonators, slotted resonators and microperforated resonators. In this thesis, perforated panel resonators are used, and the calculations for them are done with WinFlag. Results are obtained in octave-bands from 125–4,000 Hz frequency range. All absorption coefficients can be calculated for a predefined incidence angle or for a diffuse sound field, according to the standard ISO 354.

# 5 Acoustic Design of a Food Court Area in a Shopping Center

In this section the space to be treated is introduced and different solutions based on the computer simulations for the reverberation time problem is presented. Also the measurements of reverberation time at the existing side of Iso-Omena is presented.

## 5.1 Reverberation Time Measurements at Iso-Omena

In order to have some comparison and a starting point for the acoustic design of the food court area, the reverberation time at the existing side of the center aisle of Iso-Omena shopping center was measured. The existing side has some similarities compared to the new side, having a glass ceiling, hard floor and some shop windows that all reflect sound very well in the frequency range of speech. At the existing side there are, however, somewhat more wall area that is covered with wooden perforated panel resonators, and two different cavity sizes for tuning the resonant frequencies are used in the resonators. In the ceiling there is also some absorptive material at the sides of the glass ceiling. In addition, the shops are closed with a steel net walls that leak sound to the shops that have absorbing material. Also the total volume of the space is much larger than the food court area at the new side.

### 5.1.1 Measurement Tools

Measurements were done using a Norsonic Nor140 precision sound level meter, which fulfills the standards IEC 61672-1 and IEC 61260 class 1 requirements. The microphone used with the sound level meter was Norsonic Nor1229 half-inch microphone with a Norsonic Nor1209 preamplifier. The active loudspeaker used in the measurements was Powerdynamics PDE-12A. Before the measurements the sound level meter was calibrated using a standard sound source Ono-Sokki SC-2120, which gives a 94.0 dB sound pressure level at 1,000 Hz frequency.

### 5.1.2 Measurement Procedure

The measurement procedure was cut off noise, which is presented in Section 3.2, in which pink noise is played from a loudspeaker at a high level, and then cut off. An integrated measurement system found in the sound level meter was used. The calculation algorithm is in compliance with the standard ISO 354:2003. The sound level meter was installed on a stand at a height of 1.5 meters. From the sound level meter an audio cable was connected to the amplifier in the loudspeaker to provide the input signal. With only one loudspeaker a diffuse sound field cannot be created to the whole space. However, for the usability of the measurement results, the essential part is to know the reverberation time caused from sources that are near the receiver. The computer simulations are also done with similar source-receiver positioning.

When the whole setup was ready, the measurements were started. The pink noise was played and sound field was recorded the whole time and a reverberation time is calculated from the point when the source signal was muted and sound pressure level decreased from -5 dB to -25 dB. This gives the reverberation time noted with $T_{20}$. The reverberation time was measured at octave bands ranging from 125 – 4,000 Hz. A total of five different combinations of the microphone-loudspeaker pairs were measured. Measurements were conducted 14.10.2013 after the closing time of the shops in the center so that only a few people were in the space at the time of measurements. A picture from the measurement setup is presented in Fig. 14.

Figure 14: A picture of the reverberation time measurement setup performed at Iso-Omena main aisle.

### 5.1.3 Results From the Measurements

The results from the measurements are presented in Table 2. Overall the reverberation time is under 2 seconds. This is achieved mainly by having perforated panel resonators installed in the walls and the entrances to shops are closed with a steel net walls instead of movable glass walls. Also because the volume of the space is so large that air attenuation affects the sound waves at frequencies above 2,000 Hz and no disturbing room modes are formed at the audible frequency range. The frequency range where the reverberation time is the highest is at the 500 – 2,000

Table 2: Results from reverberation time measurements presented as $T_{20}$ values in seconds.

| Measurement | 125 Hz | 250 Hz | 500 Hz | 1,000 Hz | 2,000 Hz | 4,000 Hz |
|---|---|---|---|---|---|---|
| 1 | 1.32 | 1.50 | 1.49 | 1.66 | 1.63 | 1.43 |
| 2 | 1.04 | 1.69 | 1.59 | 1.73 | 1.59 | 1.31 |
| 3 | 1.06 | 1.65 | 1.80 | 1.83 | 1.73 | 1.45 |
| 4 | 1.24 | 1.69 | 1.76 | 1.72 | 1.72 | 1.43 |
| 5 | 1.02 | 1.22 | 1.61 | 1.84 | 1.67 | 1.23 |
| Average | 1.14 | 1.55 | 1.65 | 1.76 | 1.67 | 1.37 |

Hz octave bands. In the previous studies presented in Section 4.2, the reverberation time at 500 Hz in a similar space was over 2 seconds, so in comparison to that the reverberation time at the Iso-Omena main aisle can be considered excellent.

## 5.2 Basic Design

Here is presented the layout and cross-section of the food court area, available surface materials before any acoustical treatment and the results from the acoustic simulation when the food court area is left unaltered.

### 5.2.1 Layout and Surface Materials

In the extension and renovation of the Iso-Omena shopping center a food court area is designed inside the building. It is an area of about 720 square meters and with three floors the height of the space is between 19–21 meters because the ceiling is slightly raked. This can be seen in the sectional drawing presented in Fig. 15. The total volume of the space is approximately 17,000 cubic meters, including one hallway that is connected to the main atrium. The area consists of restaurant serving areas and a pathway from outside to the other parts of the shopping center. In the space there are also two trees planted on separate flower beds and an escalator from the ground floor to the second floor.

The side walls consists mainly of three large walls named as A2, B1 and C1, as depicted in Figure 16. All these walls have large amount of glass in their surfaces because they serve as the facades of different shops and restaurants in the shopping center. Also the highest part of the wall B1 consists of Paroc Acoustic 100 mm elements. Between the windows there are open surfaces that can be used to install acoustic materials. There are two material choices for the open area. Either a 5 mm thick sheet metal or 10 mm thick cement fiber board. In both cases there is a 150 mm air cavity behind the material that is created by the frames of the partition walls. Also the A2 wall is made of bricks and the architects wish is that this surface would be left untreated, or that the appearence of the wall would not change drastically.

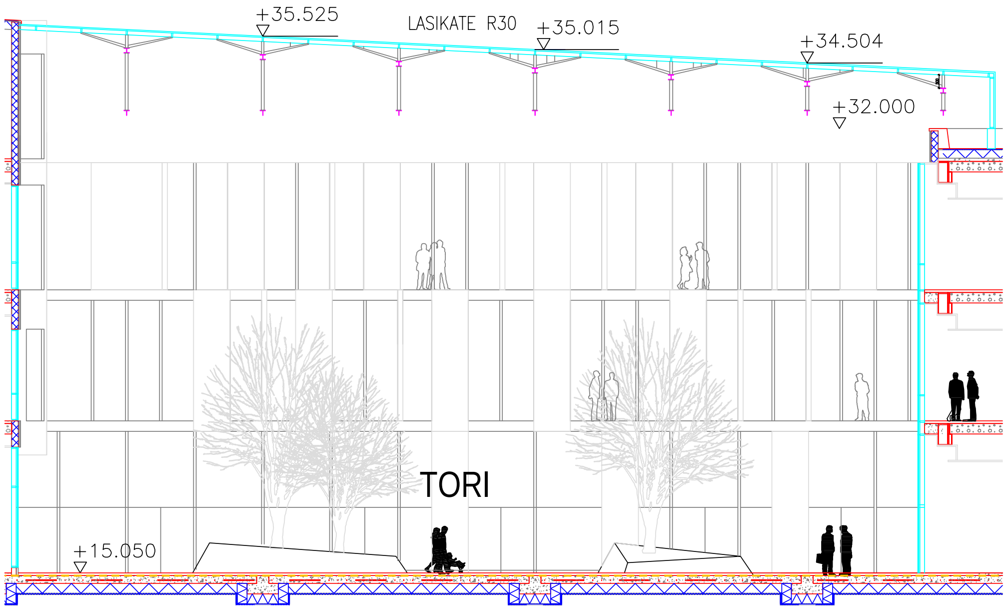


Figure 15: The sectional drawing of the food court area.

### 5.2.2 3D Simulation Method

From the drawings a 3D model was constructed using CATT Acoustic v. 9.0. A picture of the model is presented in Fig. 17. The red dots marked as A1–A3 in the figure represent omnidirectional sources. A0 is placed on the pathway to represent a walking customer and sources A1 and A2 are placed at the restaurant serving areas to represent waiters and dining customers. Sources A0 and A2 were placed at a height of 1.7 meters and source A1 at 1.1 meters height. The blue dots marked 01–04 in the figure are receivers which are placed similarly as the sources to represent different users of the space. Receivers 1, 3 and 4 were placed at a 1.7 meter height and receiver 2 at 1.7 meters. These heights were chosen to represent the average height of a Finnish person, according to the national health survey conducted in 2007 (Peltonen et al., 2008), where the average height for men was found to be 177 cm and 163 cm for women. In the figure a darker color at the surfaces indicates a higher absorption coefficient whereas lighter indicates a low absorption for that part of the wall.

The architectural design viewpoint is that the area should be like an outdoor space, and therefore the ceiling is entirely made out of glass. Floor of the space is made of concrete. In order for the sound scape to represent an outdoor area, there should be as little reverberation as possible. Given that most of the planned surface materials in the space are hard and acoustically highly reflective, the expectation is that the reverberation time of the space will be long. Since the ceiling is glass, installing porous absorptive acoustic material into it would block light coming from

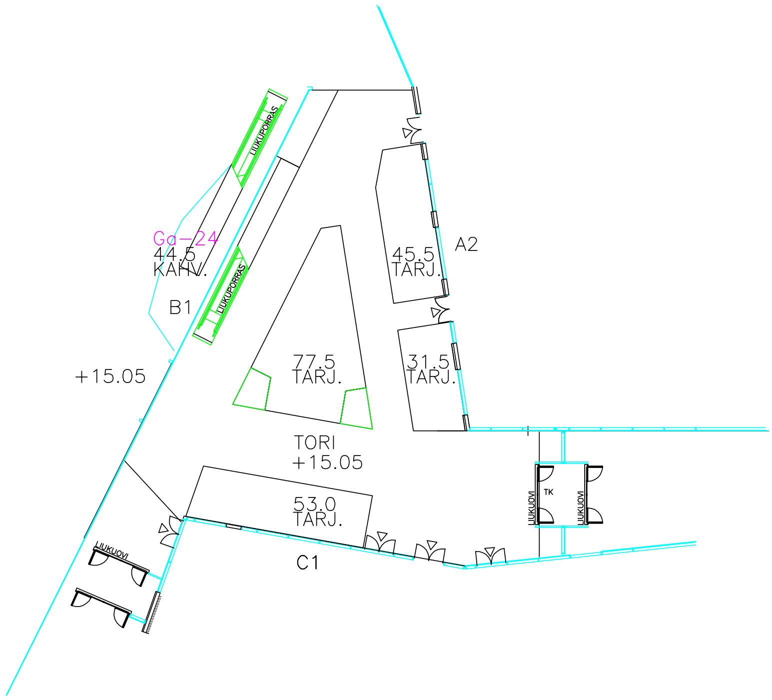


Figure 16: The floor plan of the food court area.

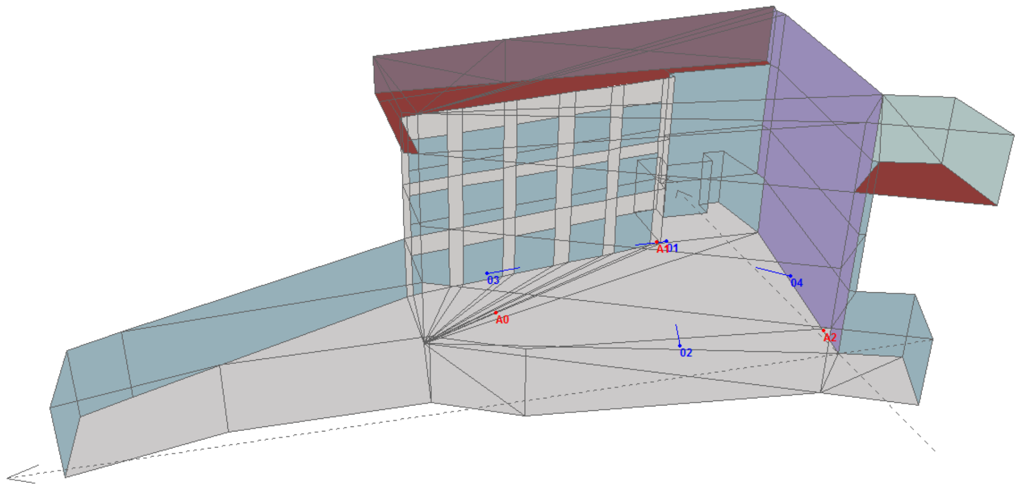

Figure 17: The 3D-model of the space made with CATT Acoustic.

the window, and only possibility would be to use microperforated panel absorbers presented in Section 2.3.5, which can be made transparent. However, this design was abandoned because over time they collect dust which reduces their absorption abilities, and their cleaning would be very difficult. Also they would have to be installed below the water sprinklers creating a fire hazard and therefore it was not feasible to use microperforated panel absorbers in this space. Therefore the three large wall surfaces are the main surfaces that can be used in the acoustic design. However, because of the limit given for the surface A2, only two surfaces remain where different installations of acoustic materials can be made.

The absorption coefficients used for the solid parts in the calculations are presented in Table 3. These include glass, concrete, 50 mm thick mineral wool, brick and Paroc Acoustic 100 mm element. The absorption coefficients are obtained from a publication of the Finnish Association of Civil Engineering (RIL 129), except for the Paroc Acoustic 100 mm, which absorption coefficient is obtained from Paroc. Absorption coefficients are presented for a diffuse sound field, where 0 means no absorption and 1 complete absorption of sound energy. From the table it is seen that both concrete and brick absorb sound very little throughout the frequency range of 125–4,000 Hz. Glass absorbs some amount of sound at the lower frequencies (125–250 Hz), but as the frequency increases, the absorption coefficient decreases. 50 mm thick mineral wool has good absorption at higher frequencies, but because of the thickness of the material, absorption coefficient starts to decrease below 500 Hz. The Paroc Acoustic 100 mm panel has reasonably good absorption throughout the frequency range.

Table 3: Absorption coefficients of the materials used in the modeling of the space at octave bands ranging from 125 Hz to 4,000 Hz.

| Material | 125 Hz | 250 Hz | 500 Hz | 1,000 Hz | 2,000 Hz | 4,000 Hz |
|---|---|---|---|---|---|---|
| Concrete | 0.01 | 0.01 | 0.02 | 0.02 | 0.02 | 0.03 |
| Glass | 0.40 | 0.30 | 0.20 | 0.17 | 0.15 | 0.10 |
| Brick | 0.02 | 0.02 | 0.03 | 0.04 | 0.05 | 0.05 |
| Mineral wool (50 mm) | 0.30 | 0.52 | 0.92 | 0.96 | 0.96 | 0.96 |
| Paroc Acoustic (100 mm) | 0.25 | 0.72 | 0.69 | 0.64 | 0.60 | 0.58 |

### 5.2.3 Results of the Basic Design

These results are obtained when the acoustical treatment is done only to the most fundamental places. This includes a 50 mm of mineral wool installed at the high platform near the glass ceiling, a 40 mm mineral wool at the ceiling of the entrance between the walls A2 and C1 and Paroc Acoustic 100 mm panel near the ceiling in walls A2 and B1, which was included in the drawings obtained from the architect. A calculation with the Sabine Equation (Eq. 23) proposes a reverberation time of 2.21 seconds at 500 Hz. Respectively, Eyring Equation (Eq. 25) gives a reverberation time of 1.98 seconds at 500 Hz. Since there are some limitations in the Sabine and Eyring equations, which are discussed in Section 3.2, the calculation is also done using the 3D model created with CATT Acoustic v. 9.0. These results are presented next.

The $T_{30}$ reverberation time at 500 Hz calculated with CATT Acoustic using the ray tracing method mapped on the food court area floor is presented in Fig.

18. From the figure it is noted that the reverberation time varies mostly between 2.5–3.0 seconds. There area no abrupt changes in the reverberation time which indicates that the sound field in the space is quite uniform. This is obtained from the diagonal walls and slanted ceiling, which all reflect sound to different directions and thus strong reflections are not created to any single point in the space. It is somewhat difficult to compare these results to the calculations done with Sabine and Eyring equations or to the measurements, and therefore a source-receiver method is also used in calculating the reverberation time. Sources and receivers were placed as explained in Section 5.2.1. These receivers and sources are used in the ray tracing calculations so that all of the sources are on at the same time, and the reverberation time for 125–4,000 Hz octave bands is calculated as the average obtained at the four receiver locations.

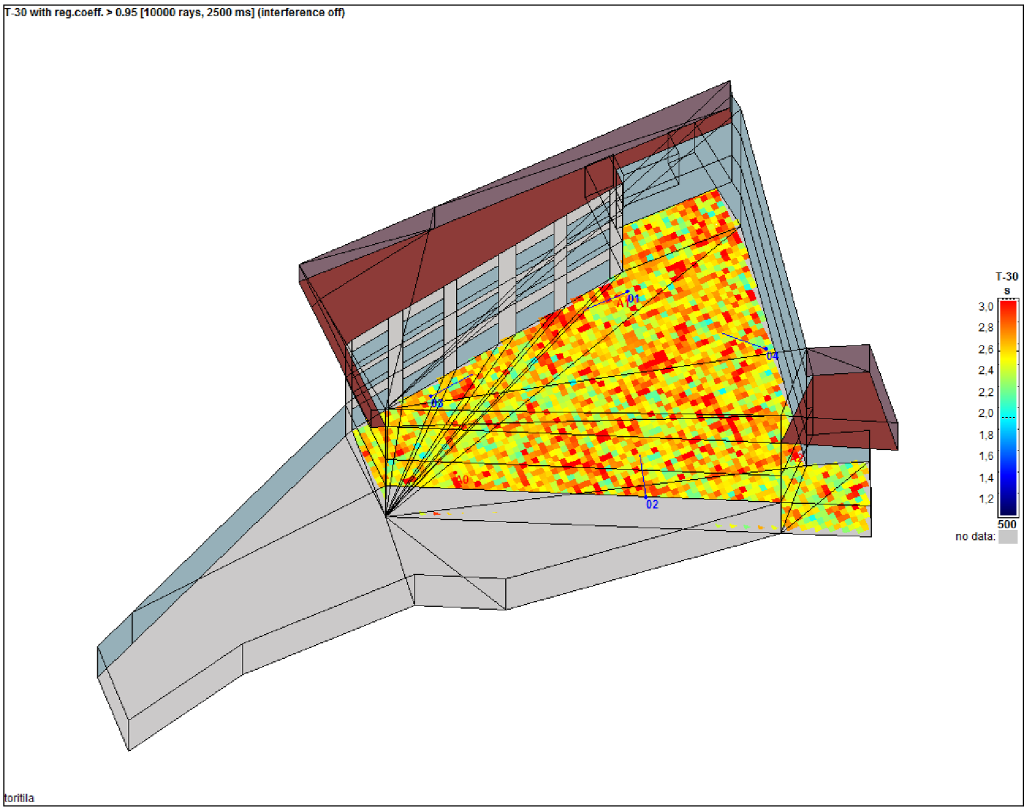


Figure 18: An illustration of the reverberation times in the whole floor area of the food court area. Red color indicates a reverberation time of 3.0 seconds and blue a reverberation time of 1.0 seconds.

Calculating with the ray tracing method using CATT, the estimated reverberation time $T_{20}$ is 2.54 seconds at 500 Hz, which is evidently higher than the results from the simple calculations. This difference can be explained with the known limitations of the Sabine and Eyring equations. Since most of the absorptive material is placed at the high platform near the ceiling, it is not visible from the ground floor where most of the sound sources will be. Also the Paroc Acoustic panels are near the ceiling and only a small part of the direct sound will reach it. Therefore the amount of direct sound from the sources that is incident to the absorptive material is low and the effective absorption area is smaller than what the physical area would suggest. Consequently the Sabine and Eyring equations gives lower reverberation times than what the ray tracing method gives. Also the Eyring equation gives even lower values, because it was designed to suit a space with high amount of absorption. Because one key requirement for the Sabine and Eyring equations was that the absorptive material should be placed uniformly into the space, it can be assumed that the results from the ray tracing method are more accurate.

A plot of the modeled reverberation time together with the measured reverberation time of the main isle at Iso-Omena is presented in Fig 19. It can be seen that the reverberation time at the food court area is over 2 seconds aboce 250 Hz and much higher than at the old side of Iso-Omena throughout the whole frequency range without any acoustical treatment. Therefore not only the impression of an outdoor space is lost, but the reverberation time is so high that the whole space becomes unpleasent for eating dinner and having a conversation becomes more difficult. This is the starting point of the acoustic design, and the goal is to find a solution that produces a reverberation time that would be between the 1.3 seconds for office lobby and 1.9 seconds for school gym at 500 Hz and to obtain significant reduction for the whole frequency bandwidth also.

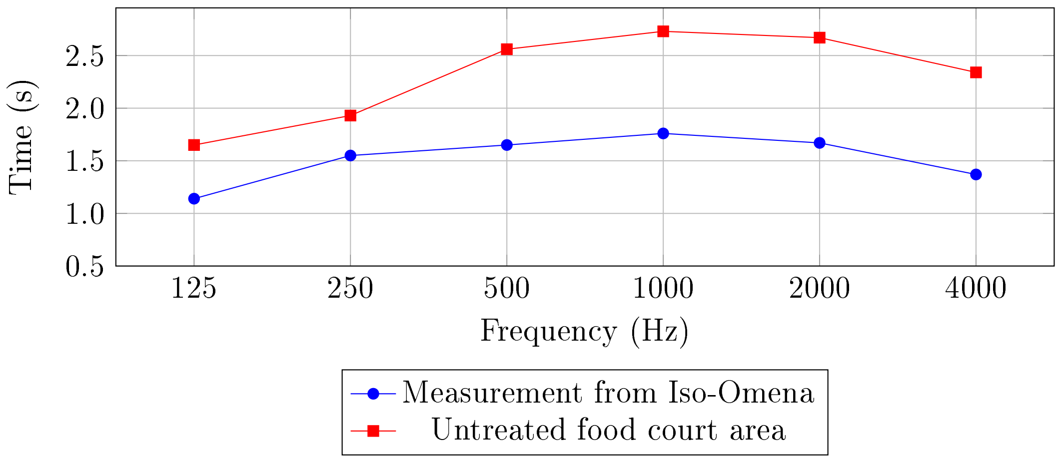


Figure 19: Reverberation time of the basic design without any additional acoustical treatment.

## 5.3 Alternative 1: Cement Fiberboard

Here is presented the first solution to the acoustic design of the food court area, using the 10 mm cement fiberboard as the material for the available walls. This material is planned to be perforated in order to create a perforated panel resonator. Calculations for different constructions of the resonators were made and results for these are also presented.

### 5.3.1 Design of the Resonator

The parameters that were used for the materials in all calculations of the resonators are presented here. Mineral wool had a flow resistivity of 10,000 sPa/m$^2$ and porosity of 95 %. The cement fiberboard panels used had a density of 1900 kg/m$^3$ and gypsum board a density of 720 kg/m$^3$. Material density for the cement fiberboard was obtained from Swisspearl (2013) and gypsum board from Saint-Gobain (2013). Air had the same properties as explained in Section 4.3.1.

To start the acoustical designing of the space, it is essential to know the available area for which acoustical treatment is possible to do. As stated in Section 5.2.1, there are two large walls where it is possible to make acoustical adjustments. However, both these walls have glass windows which has to be excluded in the area calculations. Since it is not feasible to make the acoustical 3D-model so accurate that all different windows are modeled individually, some other methods has to be used in order to employ correct absorption coefficients into the surfaces. Therefore the ratio of window area and all other surface area was calculated for the two large walls noted as B1 and C1 in the space. The windows in A2 wall were modeled accurately in order to investigate the possible effects in the reverberation time if the windows would be opened. However, this approach was abandoned because at the third floor there is a library which requires quiet environment. If the windows would be open, too much noise from the food court area would leak into the library. Using the area ratios it is possible to calculate the correct absorption coeffient for the whole wall, which consists of two different materials that have different absorption coefficients. The ratios were calculated from the architects drawings, and for the B1 wall the percentage of glass was 79 % and for C1 51 %. These were then used to calculate the weighted absorption coefficients for the whole walls.

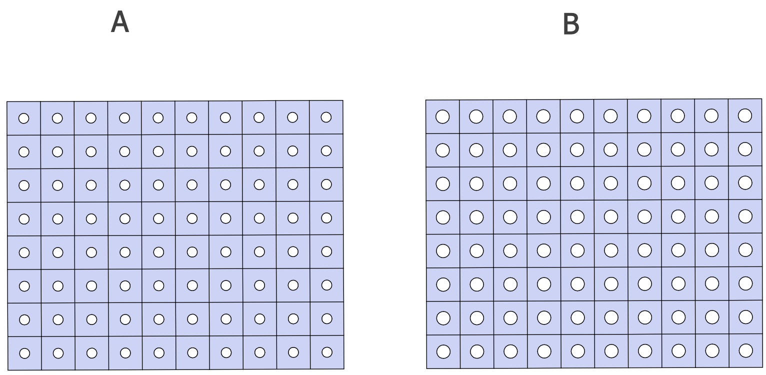


Figure 20: A: Perforations with 7 % open space. B: Perforations with 13 % open space.

Since the open wall area where it is possible to install acoustical treatment has a 150 mm cavity behind it, it is possible to design a resonator system as described in Section 2.3.4 into these walls. This is an appealing approach since it will not change the appearance of the surface, but it improves its absorption characteristics significantly. From the two possible surface materials in this alternative the case where the surface is 10 mm thick cement fiberboard is considered. The 150 mm open cavity behind the panel gives a freedom to test different combinations and thicknesses of absorptive material and air cavity behind the perforated cement fiberboard. The perforation percentage of the fiberboard was chosen to limit at maximum of 7 %, because larger open area created by the perforations would start to alter the appearance of the panel. An illustration of this is given in Fig. 20, from which can be seen that with 7 % of perforations (Fig. 20 A) the panel is still quite solid, whereas with 13 % of perforations (Fig. 20 B), the panel starts to look more hollow. Also there is a limit that the diameter of the perforations has to be small compared to the area if there are no separating elements between the holes. This imposes some difficulties to the designing, because in order to tune the perforated panel resonator to a higher frequency, the perforation percentage has to be higher, as stated in Section 2.3.4.

With the limitations explained above, three different structures for the resonator were considered. The first structure had perforated panel at the surface, behind it was the mineral wool and behind that air cavity before the solid wall. Second had a stiff panel fastened behind the mineral wool in order to limit the cavity size in order to be able to tune the resonator to higher frequencies. Third structure was that the stiff panel was also perforated and tuned so that two resonance peaks would be obtained. In order to find best possible absorption characteristics for the resonator, different combinations of mineral wool and air cavity thicknesses were tried out in each of the resonator structures. The desirable tuning frequency was chosen to be between 250–500 Hz, meaning that maximum absorption will happen at that frequency. The reason for this is that 500 Hz is the most common frequency in which reverberation time is investigated. Also in the same frequency region is the frequency content of speech (Karjalainen, 2009; Rossing et al., 2002), and since speech will be the dominant sound in the space it is feasible to try to attenuate it as much as possible. Different solutions were calculated using the WinFlag program.

In Fig. 21 are plotted the absorption coefficients for a resonator system consisting of a perforated panel backed by a mineral wool and air cavity. All absorption coefficients are calculated for a diffuse sound field. Three different mineral wool thicknesses were modeled. Because of the limit in the perforation percentage, the resonance could not be tuned to exactly 500 Hz. This is because the cavity behind the panel is fixed at 150 mm in depth, and from Eq. 14 in Section 2.3.4 it was seen that when the air cavity increases the perforation percentage has to increase also in order to get the resonance to a higher frequency. Also adding mineral wool to the cavity further lowers the resonance peak. Considering this, with 7 % of perforations the theoretical resonance peak happens roughly at 350 Hz. From Fig. 21, it is seen that when the thickness of absorptive material is increased behind the perforations, the resonance peak widens and moves to lower frequency, as was stated that would happen in Section 2.3.2. In spite of the resonance peak not being directly at 500 Hz, a reasonable absorption is still obtained at the frequency, especially when the mineral wool thickness is 100 mm or 140 mm. In consequence, the absorption at 250 Hz is also very good. After 500 Hz the absorption cofficient starts to drop and over 1,000 Hz frequencies there are no difference whether the mineral wool layer is

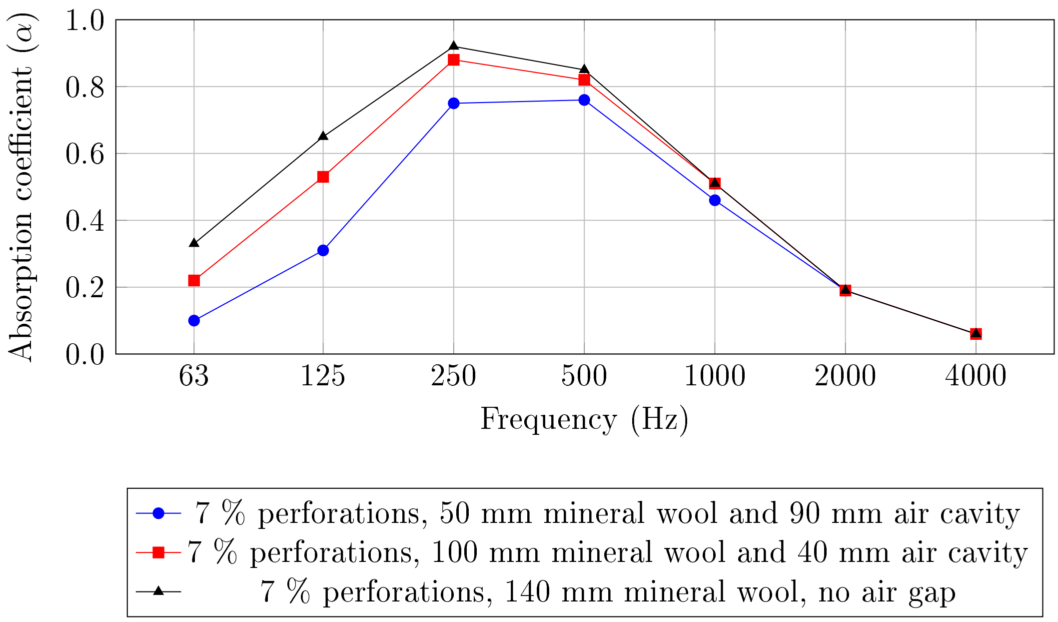


Figure 21: Absorption coefficients of resonator with perforated panel, mineral wool and air cavity with varying mineral wool thickness.

50 mm or 140 mm thick. This is actually a good result, because it can be assumed that materials in the space, including chairs and tables that will be in the area will provide absorption at higher frequencies, but not so much at frequencies below 1,000 Hz.

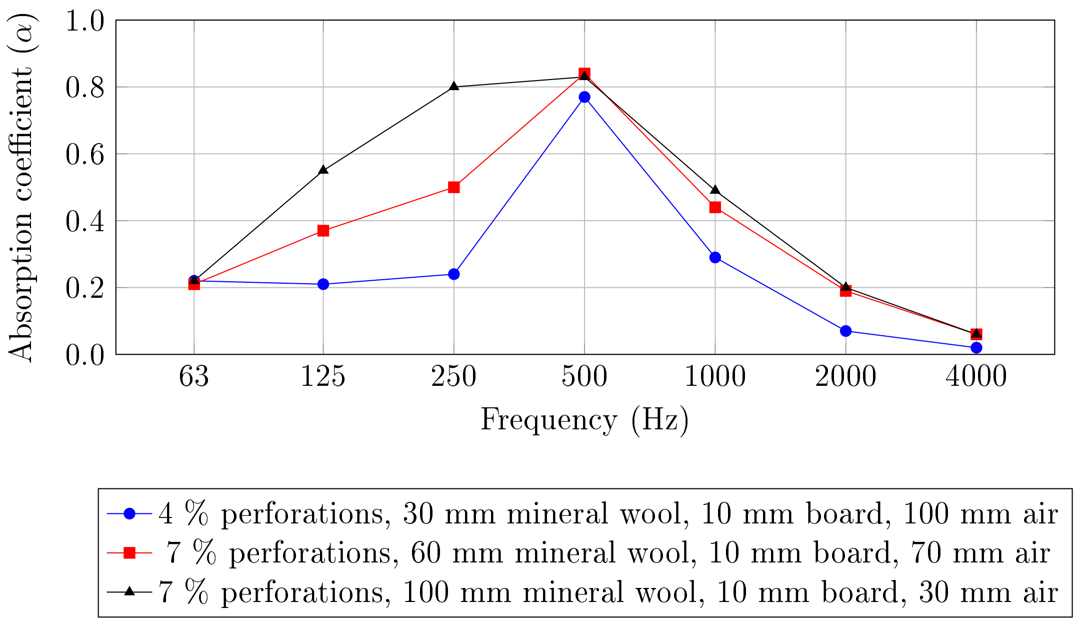


Figure 22: Absorption coefficients of resonator with panel behind the mineral wool.

The second resonator structure consisted of a resonator, mineral wool layer, solid panel behind it and an air cavity before the solid wall. The idea of this design was that by inserting a solid panel behind the resonator the air cavity is made smaller. In consequence, the resonance peak could be tuned to 500 Hz. The solid panel behind the mineral wool was chosen to be ordinary gypsum board. Again, three different thicknesses for the mineral wool were modeled, as shown in Fig. 22. The thinnest mineral wool was 30 mm and thickest 100 mm, because if the thickness would grow much from that the resonator would start to resemble the same case as was considered in the previous paragraph. The panel behind the mineral wool allows the resonator to be tuned at 500 Hz with the 7 % perforations limitation. Likewise the previous case, when the mineral wool is increased, the resonance peak widens. This approach does not give better absorption coefficients than without the panel behind the mineral wool, and since it is more complex in design, this structure will not be used.

Third structure was the same as the second, except that now the gypsum board behind the mineral wool was also perforated, creating a double resonator. These two resonators can be tuned individually to obtain two resonance peaks and thus widening the absorption bandwidth. Since this construction is more complex than the first two, results are calculated for four different combinations of mineral wool thicknesses. Results from the modeling are presented in Fig. 23. From the different mineral wool thicknesses it is seen that making the layer thicker than 50 mm won't enhance the resonance peak at 500 Hz. Only the lower resonance is emphasized, but even there the 80 mm mineral wool layer produces the best result, and after that the improvement is only marginal.

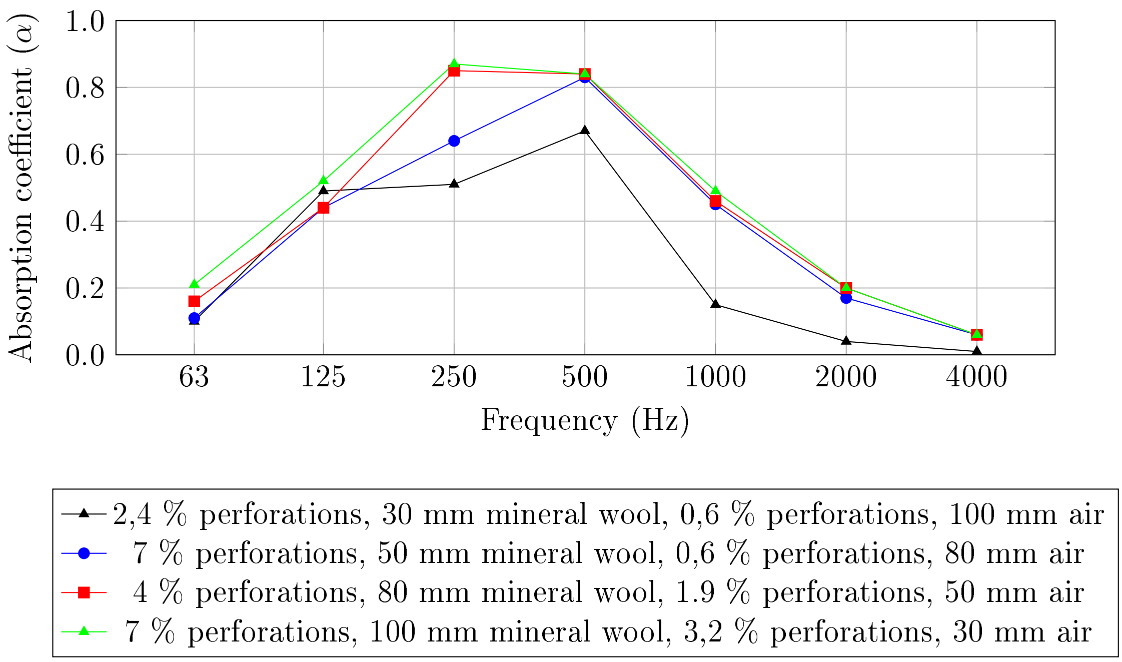


Figure 23: Absorption coefficients of resonator with perforated panel behind the mineral wool.

### 5.3.2 Results of Alternative 1

From the different resonator designs presented in the previous section the best option should be used. If the only factor in choosing the resonator is absorption performance, the best option would be 7 % perforations backed with 140 mm of mineral wool. However, since the mineral wool thickness has more effect on the absorption coefficient in the 250 Hz octave band, thinner layer is also acceptable. The different systems presented in Figures 22 and 23 provide also a bit better absorption coefficients. However, the improvement is less than 0.1 in the 500 Hz octave band and because of their more complex design they are more susceptible to installation errors which could be detrimental to their absorption abilities. On these grounds, the final resonator system was chosen to be panel with 7 % of perforations with 50 mm mineral wool and 90 mm air cavity behind it.

In these results, the only change in the materials compared to the design presented in Section 5.2.3 is that the surface materials in the walls B1 and C1 are perforated to create a resonator defined in the previous section. The absorption coefficients for the whole wall surfaces are obtained with the method described in the previous section, using the area weighting. Calculations with the 3D-model were done with the same receiver and source locations than with the basic design calculations. Using the Sabine equation to calculate the reverberation time gives 1.95 and with Eyring equation 1.72 seconds at 500 Hz. Calculations using the 3D-model and ray tracing gives a reverberation time $T_{20}$ of 2.07 seconds. In Fig. 24 are plotted the measured reverberation time at Iso-Omena and the results from the ray tracing modeling at 125–4,000 Hz octave bands. With this design the reverberation time is under 2 seconds at 250 Hz and just over 2 seconds at 500 Hz. In Fig. 25 is presented the mapped reverberation time at 500 Hz for the whole food court floor area. Compared to the case where no absorptive material was installed, the reverberation time has dropped below 2.2 seconds for nearly every point in the area.

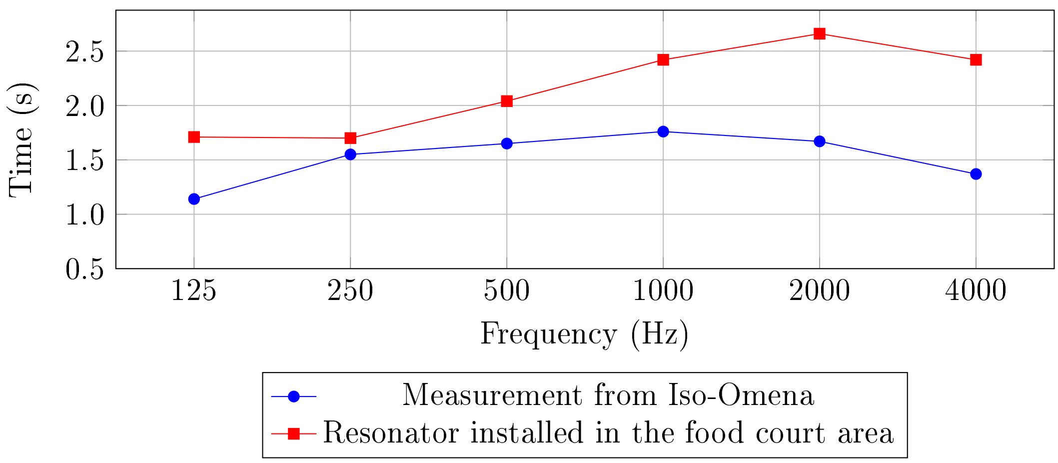


Figure 24: Reverberation time of the basic design with resonators installed at walls B1 and C1.

Again the reverberation times that were calculated with Sabine and Eyring equations are lower than the reverberation time obtained with the ray tracing method.

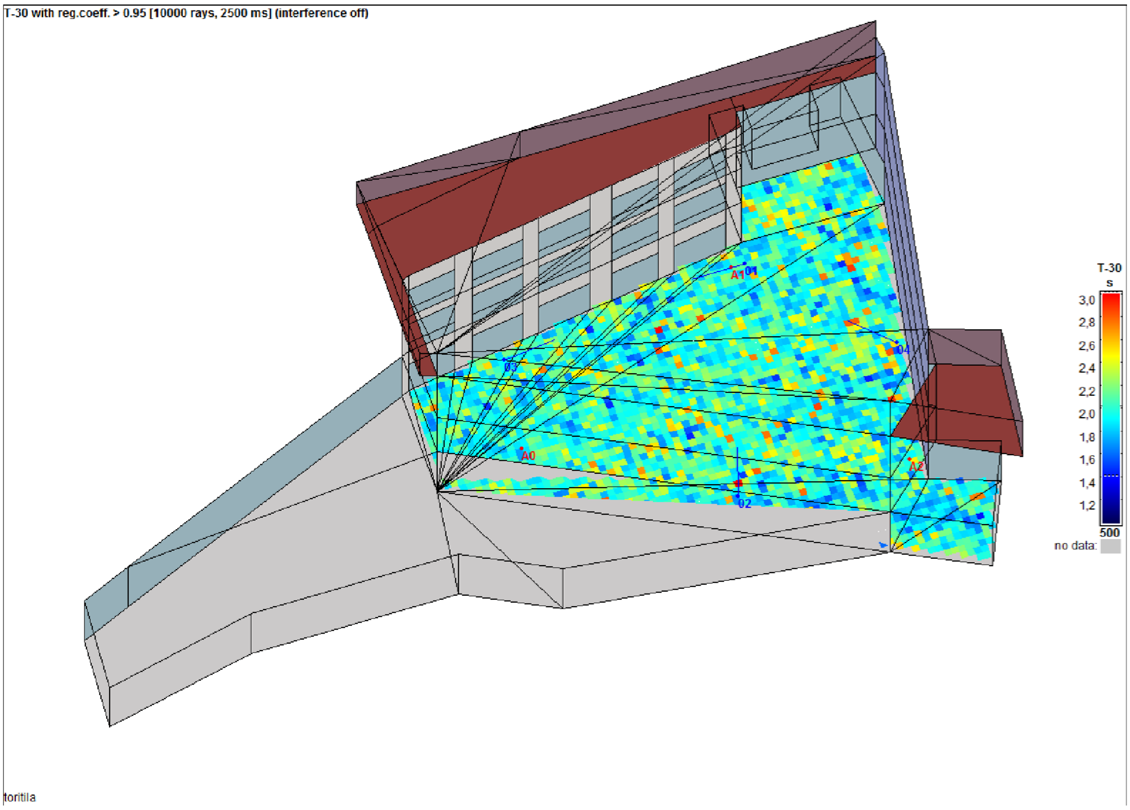


Figure 25: An illustration of the reverberation times at 500 Hz for the whole floor area of the food court area. Red color indicates a reverberation time of 3.0 seconds and blue a reverberation time of 1.0 seconds

In the literature it is often noted that Sabine equation gives higher reverberation times than what is measured in practice, as discussed in Section 3.2. However, now the Sabine reverberation time is much closer to the $T_{20}$ value than with the calculations of the first model. Also the Eyring reverberation time is a bit closer, the difference being 0.35 seconds compared to a nearly half second which was the case with the basic model. The reason for this is that there are significantly more absorptive material in the space now, and it is also distributed to a much larger area. This fulfills better the requirement that the absorptive material has to be evenly distributed in the space in order for the Equations 23 and 25 to give correct answers. Overall this quite simple alteration to the surface materials has provided a significant improvement to the acoustic conditions of the space, without changing the appearance of the surface too much.

The goal was to get the reverberation time preferably under 1.9 seconds at 500 Hz so this approach does not straightforwardly fulfill this requirement. However, as was stated in Section 4.2, the computer simulations have a tendency to overestimate the results compared to the measurements done in a same space. The difference was reported to be about 10 %, and if that is taken into account here, the reverberation time at 500 Hz would be about 1.8 seconds. This is in between of the reverberation time guidelines for an office lobby and school gym given in the Finnish building regulations collection RakMK C1-1998.

## 5.4 Alternative 2: Sheet Metal

Here is presented a second solution to the acoustic design of the food court area. Now, the surface material for the B1 and C1 walls is 5 mm thick sheet metal. The same approach is undertaken as with the alternative 1, and different perforated resonator systems are modeled in order to find a best possible solution. The best option is then chosen and reverberation time simulations done with the chosen resonator installed at the walls.

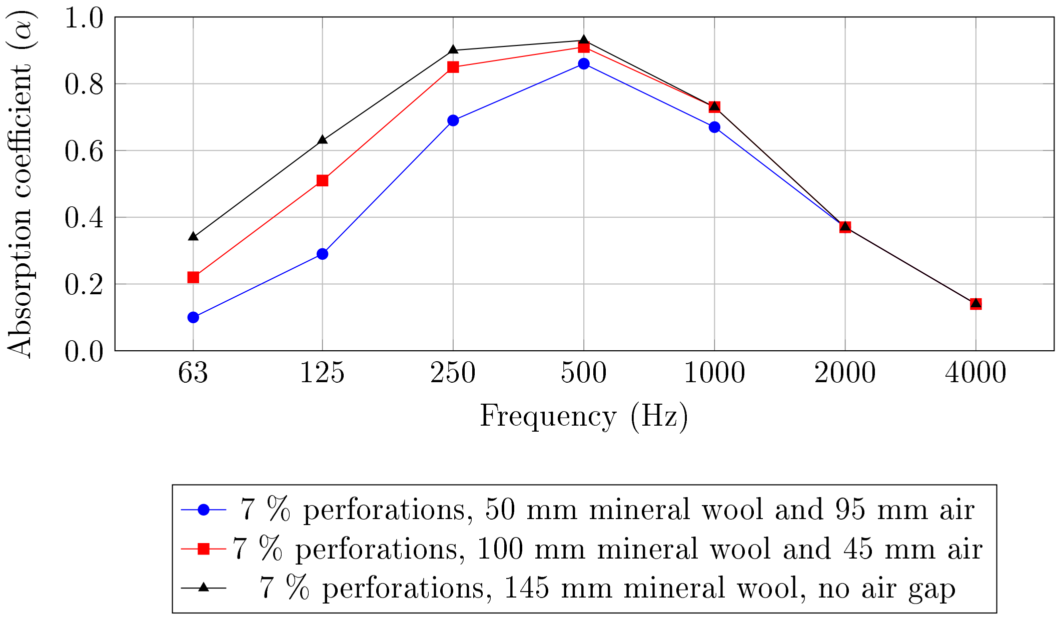


Figure 26: Absorption coefficients of resonator with perforated panel, mineral wool and air cavity with varying mineral wool thickness. Perforated panel is a 5 mm thick sheet metal.

### 5.4.1 Design of the Resonator

The same material parameters for the resonator design was used as with the alternative 1, except that the density of the 5 mm sheet metal was 7000 kg/m$^3$. Also the same approach was used, which included the modeling of three different constructions for the resonators. The results for the different mineral wool thicknesses for the resonator consisting of a perforated 5 mm thick sheet metal, mineral wool layer and an air cavity is presented in Fig. 26. Now that the perforated panel is half as thick as with the alternative 1, the resonance peak shifts to a higher frequency if the perforation percentage is kept the same. It can be seen that the resonance peak is now quite close to the 500 Hz. Also the Q-factor of the system is now lower, resulting in a wider absorption bandwidth. Compared to the first alternative, the main difference is the improved absorption at higher frequencies. This is a result of a better absorption at oblique angles obtained from the thinner perforated panel.

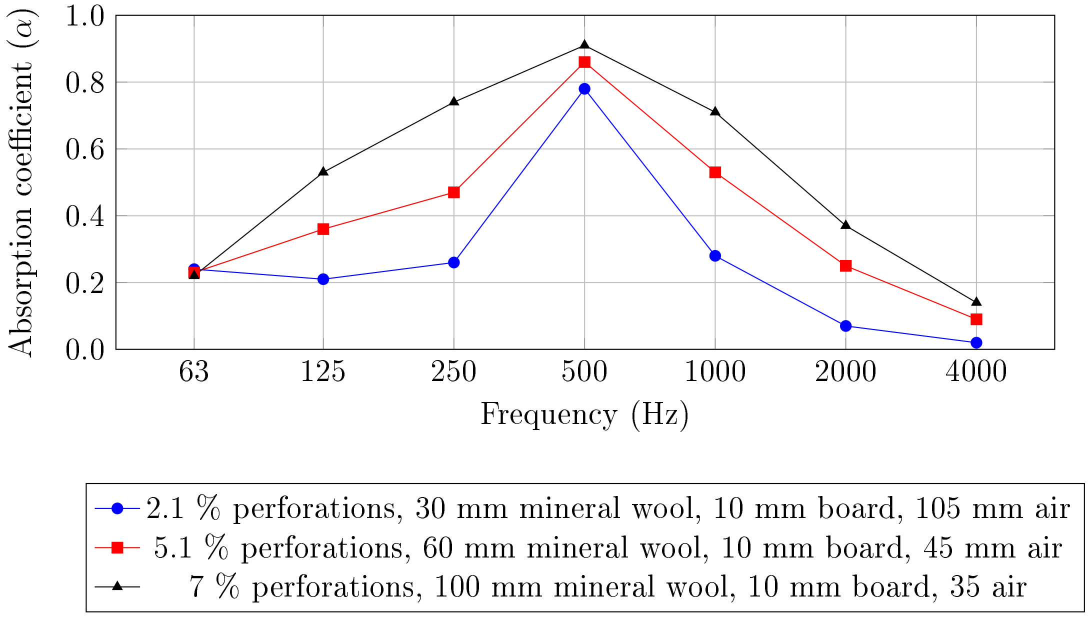


Figure 27: Octave band absorption coefficients of resonator with perforated panel, mineral wool and air cavity with varying mineral wool thickness. Perforated panel is a 5 mm thick sheet metal.

The second resonator structure was perforated panel, mineral wool layer, panel backing the mineral wool and air gap. Results from the modeling is presented in Fig 27. This design has the most definite resonance peak visible. It is observed that when more mineral wool is inserted behind the perforated panel, the resonance peak gets stronger at the 500 Hz and also broader. At first this might seem controversial to the theory presented in Section 2.3.4, as there it was stated that the resonance could be adjusted with porous material to be either high and narrow or lower but broader. There is a rational explanation to the phenomenon seen here. With thinner amount of mineral wool, the resonance peak is stronger in the tuning frequency. Since the absorption coefficients are calculated for octave bands, the neighboring one-third octave bands have also significant effect to the absorption coefficients. Therefore when the resonance peak slightly lowers but also broadens, the absorption at the nearest one-third octave bands is increased, resulting in overall higher absorption coefficient at the 500 Hz octave band.

The third structure was a perforated panel, mineral wool, another perforated panel and an air gap. Again, similar modeling was performed as with the 10 mm cement fiberboard as the surface material. Results from the modeling is presented in Fig. 28. Similar behaviour is observed as with the previous two resonator structures: the absorption at 500 Hz is more emphasized and also the overall absorption is higher. The perforation percentages for the outer layer are a bit lower than with the alternative 1. This is a consequence from the thinner perforated panel, as it was stated earlier that the tuning frequency increases when the panel thickness decreases.

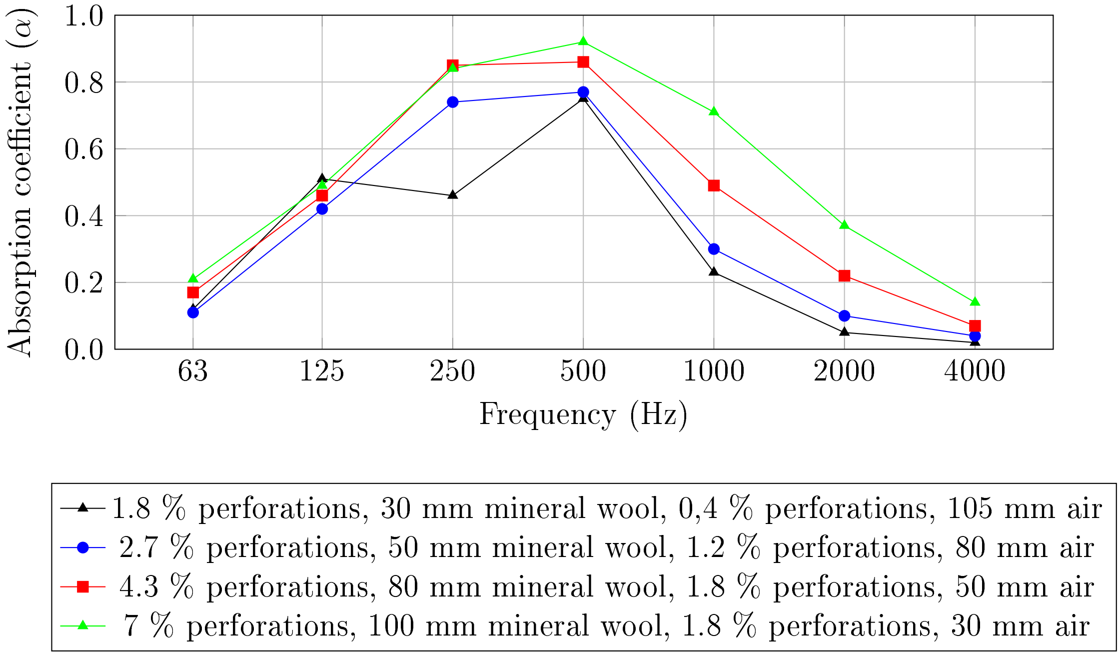


Figure 28: Absorption coefficients of resonator with perforated panel behind the mineral wool. Perforated panel 5 mm sheet metal.

### 5.4.2 Results of Alternative 2

In order to be able to choose the best alternative, from the three different designs a few constructions were chosen and the weighted absorption coefficients for the B1 and C1 walls were calculated. Sabine reverberation times were then calculated using the resonators in the walls. Results from this calculation is presented in Table 4. It is seen that there are no significant improvement from the thicker layer of mineral wool or more complex double resonator systems over the simple resonator with 50 mm mineral wool. With every design, the reverberation time is under 2 seconds below 1,000 Hz frequencies. Also the 50 mm layer of mineral wool is cheaper than having 145 mm of mineral wool.

Table 4: Reverberation times calculated with Sabine equation of the food court area with different resonators at the walls.

| | 125 Hz | 250 Hz | 500 Hz | 1,000 Hz | 2,000 Hz | 4,000 Hz |
|---|---|---|---|---|---|---|
| Resonator 1 [4] | 1.89 | 1.74 | 1.86 | 2.03 | 2.19 | 2.04 |
| Resonator 2 [5] | 1.75 | 1.66 | 1.83 | 2.00 | 2.19 | 2.04 |
| Resonator 3 [6] | 1.81 | 1.68 | 1.83 | 2.02 | 2.19 | 2.04 |

[4] 7 % perforations, 50 mm mineral wool and 95 mm air.
[5] 7 % perforations, 145 mm mineral wool.
[6] 4.3 % perforations, 80 mm mineral wool, 1.8 % perforations, 50 mm air.

Based on the discussion and results presented in the previous paragraph, the resonator for this alternative is also chosen to be the 7 % perforations backed by a 50 mm mineral wool layer. With this resonator installed at the B1 and C1 walls, the Sabine reverberation time is 1.86 and Eyring reverberation time 1.63 seconds. In comparison to the 2.01 seconds that is obtained from the simulation, the results are in line with the results obtained in Section 5.3.2. A complete plot of the simulated reverberation times are presented in Fig. 29. The clearest difference to the alternative 1 is the lower reverberation time in the 1,000–4000 Hz octave bands.

In the Fig. 30 is presented the simulated reverberation time at 500 Hz plotted to the floor of the food court area. The overall reverberation time is a little bit lower than with the alternative 1, and again there are no clearly visible any areas where the reverberation time is emhasized. Instead the reverberation time is uniform throughout the area. Again if the 10 % overestimation is assumed for the reverberation time, the actual reverberation time in the space at 500 Hz is about 1.8 seconds.

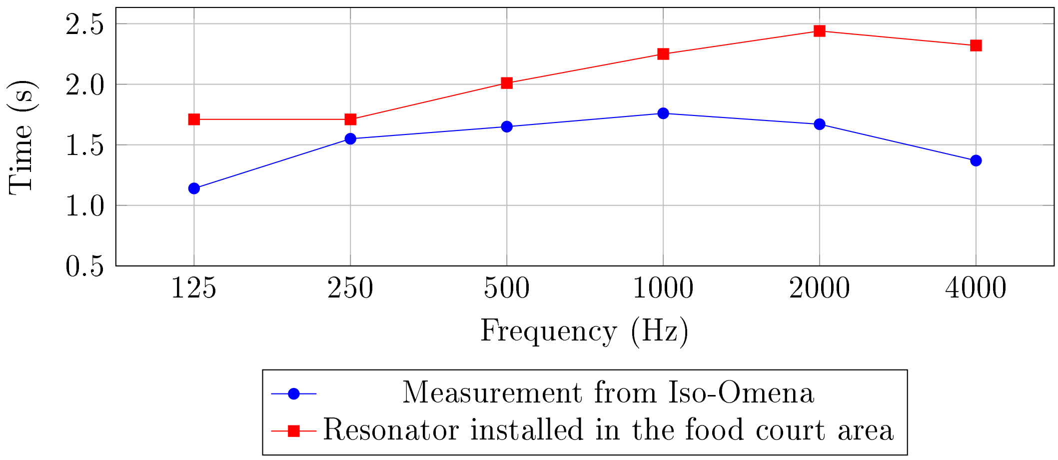


Figure 29: Simulated reverberation times of the alternative 2 with resonators installed at walls B1 and C1 plotted with the measured reverberation time at Iso-Omena.

### 5.4.3 Comparison of Alternatives 1 and 2

Overall the results of alternative 1 are quite similar to the results obtained from the alternative 1. Two differences can be noted. First, the perforation percentage is significantly smaller with the thinner perforated plate. Also there is a little bit more absorption at the higher frequency range of 1,000–4,000 Hz. However, the furnitures that will be installed into the space have an effect on the absorption at that range, and they also provide more diffusion which spread the sound energy even more evenly. Therefore the choice between the two surface materials for the walls B1 and C1 is in terms of the reverberation time insignificant, since both can be tuned so that sufficient absorption is obtained. However, if the visual appearance is more of a concern, the 5 mm sheet metal has a better starting point to be tuned

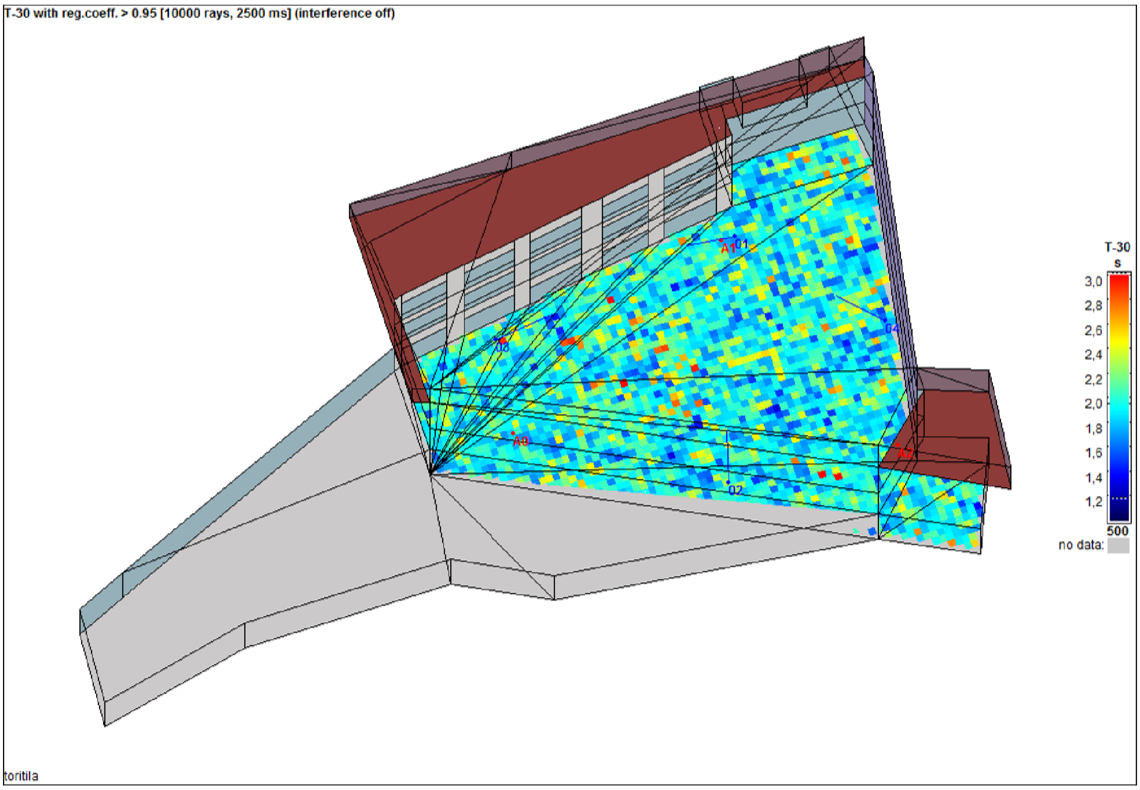


Figure 30: A plot of the simulated reverberation times at 500 Hz for the whole floor area of the food court area. Red color indicates a reverberation time of 3.0 seconds and blue a reverberation time of 1.0 seconds

with smaller percentage of perforations. This will result in smaller holes that are also more unnoticable.

From the absorption point of view, best absorption coefficient is obtained with 5 mm sheet metal with 7 % of perforations backed by 145 mm of mineral wool, i.e. the whole cavity behind the sheet metal is filled with mineral wool. However, as it was expressed previously, nearly equal absorption coefficient at 500 Hz can be obtained with thinner mineral wool layer also. Therefore the final choice for the wall material is 5 mm thick sheet metal with 7 % of perforations with 50 mm mineral wool and 95 mm air gap behind the perforated panel.

# 6 Conclusions

In this thesis a simulation of the reverberation time of a food court area at the extension of Iso-Omena shopping center located in Espoo, Finland, is presented. In the area there was acoustically highly reflective surfaces, and the reverberation time of the space would be high without any acoustical treatment. Since the ceiling and significant amount of the walls in the space was glass, there were a limited amount of surfaces in which acoustical treatments could be done. Also the appearance of the surface materials should not change dramatically. From these points of view a solution in which the reverberation time at 250–500 Hz region could be lowered was looked for. The method was a computer simulations using CATT Acoustic v. 9.0 software for the reverberation time calculations and WinFlag 2.4 for calculations of the resonators.

The easiest way of doing the acoustical treatment would be to install mineral wool to the ceiling. Because of the ceiling is glass, this design has to be abandoned. Instead, the surfaces of two walls in the space could be perforated in order to create a perforated panel resonator. This way the appearance of the surfaces would not change and yet a good absorption would be obtained at 250-500 Hz region. There were two different options for the surface materials that had different thicknesses. As the thickness of the panel in a perforated panel resonator affects the resonance frequency, these two options were investigated in order to find a good resonator structure. It was found that with both surface materials it is possible to tune the resonator so that the reverberation time in the space could be lowered significantly. The main difference between the two materials is that with the thinner surface material there are a bit more options concerning the perforations in the surface. E.g. if the smallest possible holes are preferred, then the thinner surface material would be better for this space.

Different resonator structures for both of the surface materials were modeled. The best absorption were found to be with a resonator that was made of 5 mm sheet metal, which had 7 % of perforations, and behind the panel was 145 mm of mineral wool. However, as the most important frequency bandwidth that was considered is the 500 Hz octave band, a construction of 7% perforated sheet metal backed by a 50 mm of mineral wool and 95 mm air gap provides almost as good absorption at the 500 Hz frequency. As this structure is cheaper, this is recommended as the final alternative for the wall material.

In the results it is shown that without any absorptive material in the space the simulated reverberation time of the space was 2.54 seconds at 500 Hz. With the final alternative installed at the walls the simulated reverberation time is 2.01 seconds. In the literature there are suggestions that the simulated reverberation times are about 10 % higher than measured reverberation times at the same space. Considering this the reverberation time of the space will be about 1.8 seconds. In order to further improve the accuracy of the simulations, the 3D model could have been constructed in more detail. However, this approach would consume high amount of time and since there may be some alterations to e.g. in the furnitures and other movable items in the space, it is more suitable to model accurately the parts that can be assumed to be permanent. This way a basic understanding about the acoustic characteristics of the space is obtained, and this can then be further developed, if necessary.

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